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The Marine Engineer
AND NAVAL ARCHITECT.

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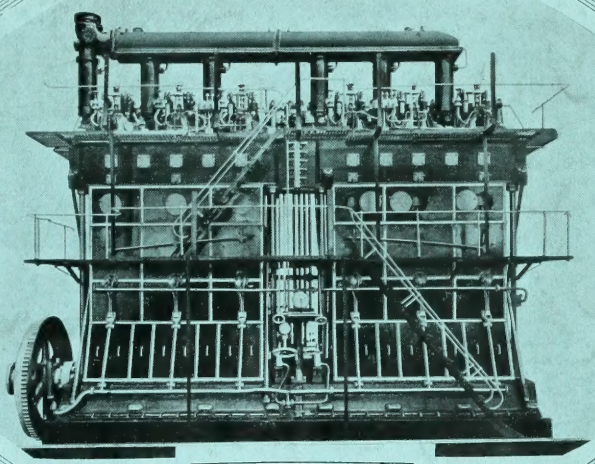

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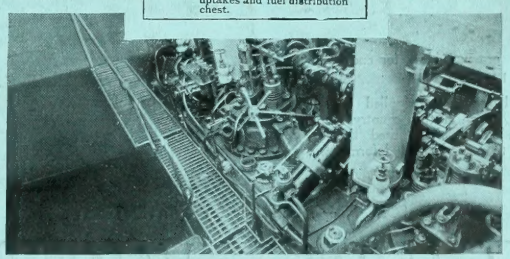
Below: Reversing Engine, exhaust uptakes and fuel distribution chest.



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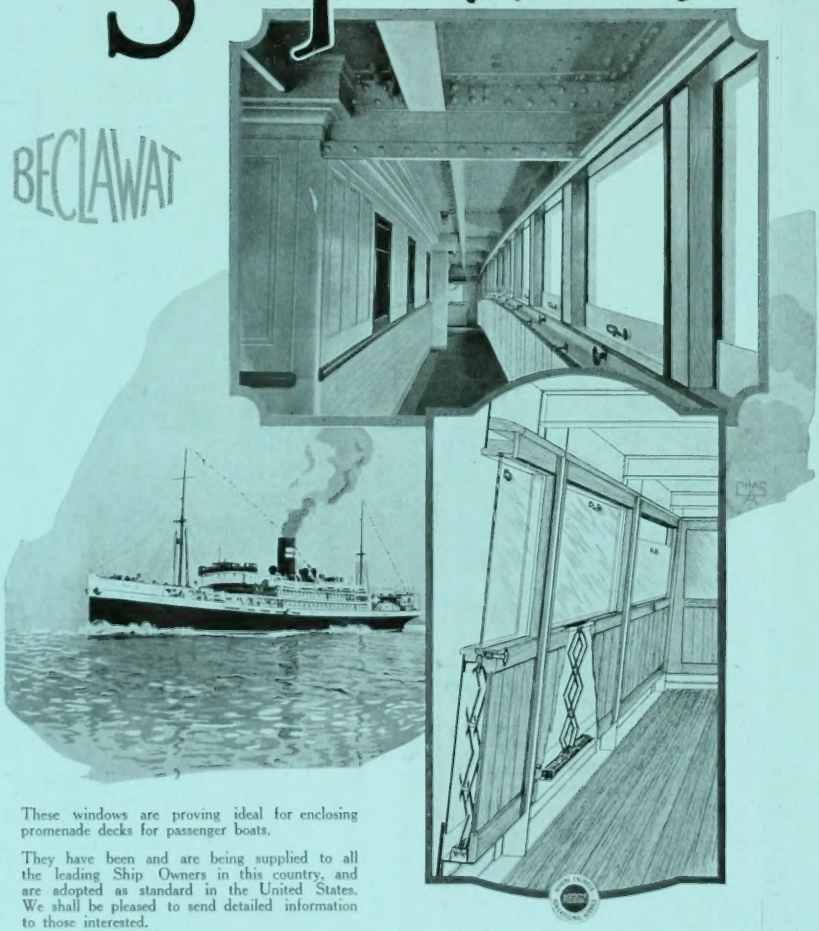

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The Marine Engineer
AND NAVAL ARCHITECT.

A Practical Journal for all Concerned in the Design Construction and Maintenance of

MECHANICALLY PROPELLED SHIPS — MARINE ENGINEERING WORKS
DIESEL AND OTHER OIL ENGINES — WELDING & MECHANICAL REPAIRS
AUXILIARIES & REFRIGERATION — CARGO HANDLING APPLIANCES

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The Editor will be glad to consider original articles and paragraphs on all subjects indicated by the title. Short practical articles based on the sender's own experience and illustrated by photographs or drawings are desired.

VOL. XLVI. No. 548. MAY, 1923.

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The funeral took place recently of Mr. H. J. Hodgson, who was a director of C. & H. Crichton Limited, ship-repairers and engineers, Liverpool.

The late Mr. Robert Lee Priestley, Marine Engineer, Mayor of Gravesend, left estate of the gross value of £5,111, the net personalty being £2,509.

At the annual meeting of the Institute of Marine Engineers, held recently, Viscount Curzon, M.P., who holds the rank of Commander in the Royal Naval Reserve, was elected President.

The death has occurred at Rotterdam of Mr. J. F. B. Schupper, at the age of 41 years. He was a Managing Director of the N. V. Furness Scheepvaart & Agentuur My., Rotterdam.

Mr. Gilbert Lacy, Foreman Engineer in the repair department of Cammell Laird & Co. Limited, Birkenhead, has relinquished his position after 30 years' connection with the firm.

The death occurred recently, at Liverpool, of Mr. James Ambrose Collard, senior partner in the firm of Collard & Turner, Consulting Marine Engineers, Water Street, Liverpool. Mr. Collard was 61 years of age.

Mr. Arthur Tarrant, the Senior Engineer Surveyor to the Southampton Office of the Board of Trade, has been promoted to the position of Principal Officer at Glasgow.

Mr. John R. R. Wilson, M.Inst.C.E., H.M. Divisional Inspector of Mines, who has died in a nursing home in Newcastle after a long illness, worked as an engineer in the construction of the Manchester Ship Canal.

In order to take up his new duties as assistant to the president of the Diamond Power Speciality Corporation, of Detroit, U.S.A., Mr. Albert W. Honywill has severed his connections with the Terry Steam Turbine Company.

Mr. R. H. Winstanley, who for a good many years has been Honorary Treasurer to the North-East Coast Institution of Engineers and Shipbuilders, has been presented by the members with a massive silver salver suitably inscribed, and a tall clock in Chippendale style, mahogany, in recognition of his services.

Mr. David Young Cassels, of Fairclough, Bothwell, formerly of Sandyhills, Shettleston, of Pott, Cassels & Williamson Ltd., engineers and ironfounders, Motherwell, and director of the Dalmellington Iron Company Ltd., and the Steel Company, of Scotland, Ltd., has left personal estate in the United Kingdom, of the value of £137,244.

Mr. Anthony John Davison celebrated his 74th birthday on April 21. Mr. Davison is the Senior Honorary Engineer-Commander Royal Naval Reserve, his commission dating from June 30, 1889. He was then Superintendent to Wm. Milburn & Co.'s Anglo-Australian Steam Navigation Co. Mr. Davison has been privileged to attend the Levées of three Sovereigns.

Mr. Joseph Jack is relinquishing his position as Engineer and Works Manager of the Parceria dos Vapores Lisbonenses (Lisbon Dry Dock Company), on account of ill-health, and will return to England in the early summer to recuperate. Mr. Jack has held the position for several years, having been actively associated with the development to its present dimensions.

Mr. G. H. Froebel has been appointed manager of the Marine Sales department of the Westinghouse Electric and Manufacturing Company. Mr. Froebel was formerly manager of the Houston, Texas, branch office of the Westinghouse Electric and Manufacturing Company, and his new headquarters will be at the East Pittsburg works of the Westinghouse Company.

Mr. H. S. Aspinall, who recently resigned his appointment as Sales Manager and Director of the Keighley Gas & Oil Engine Company Limited, has been appointed General Sales Manager to Vickers-Petters Limited, of Ipswich Works, Ipswich, manufacturers of vertical two-cycle oil engines for stationary and marine purposes. Mr. Aspinall's headquarters will be at Vickers-Petters' works at Ipswich.

Mr. William Douglass, the famous lighthouse engineer, who has died at Penzance in his ninety-third year, had very old family associations with Northumberland and also Durham. One of his earliest recollections was seeing *Puffing Billy*, which was among the earliest of the locomotives in this country, when it first commenced running in the Newcastle district. His father, Mr. Nicholas Douglass, and his brother, Sir James Douglass, both received the freedom of Newcastle.

Captain Charles W. Reed, B.Sc., A.M.I.N.A., has been appointed Ship and Engineer Surveyor to Lloyd's Register of Shipping, and will shortly take up his duties with the Society at Liverpool. Captain Reed is the son of Mr. S. W. Reed, late Engineering Manager at Palmers & Doxfords, and has lately been engaged in research work with Palmers Shipbuilding & Iron Company. He served throughout the European War, was twice mentioned in despatches and was awarded the Croix de Guerre.

EDITORIAL.

THE MARINE ENGINEERING SITUATION.

LLOYD'S Register Shipbuilding Returns for the quarter ended March 31, 1923, have been issued recently, and a perusal of the figures shows not only the trend of the shipbuilding situation as a whole, but also throws an interesting light on the percentage of motorship tonnage to the total tonnage commenced during the last four quarters. Table I shows for each of the principal shipbuilding countries the number and tonnage of steam and motorships commenced

TABLE I.—SHOWING STEAM AND MOTORSHIP TONNAGE COMMENCED DURING QUARTERS ENDING :—

Country.	March 31, 1923.		Dec. 31, 1922.		Sept. 30, 1922.		June 30, 1922.	
	No.	Tonnage.	No.	Tonnage.	No.	Tonnage.	No.	Tonnage.
United Kingdom :								
Steamers	110	303,949	64	199,307	30	66,908	12	37,177
Motor vessels	10	48,024	5	30,260	3	15,200	5	810
Denmark :								
Steamers	8	11,916	2	2,060	1	960	1	1,050
Motor vessels....	1	107	4	10,610	3	7,605	1	2,925
France :								
Steamers	9	48,830	2	14,800	2	5,800	11	23,360
Motor vessels....	—	—	9	3,900	—	—	—	—
Holland :								
Steamers	2	2,900	5	4,400	—	—	3	12,100
Motor vessels....	2	360	—	—	—	—	1	5,250
Italy :								
Steamers	—	—	2	7,300	—	—	2	200
Motor vessels....	—	—	—	—	—	—	—	—
Japan :								
Steamers	5	5,530	12	12,234	10	16,120	3	13,990
Motor vessels....	—	—	—	—	—	—	—	—
Norway :								
Steamers	10	5,145	3	4,744	5	6,689	3	3,389
Motor vessels....	4	725	—	—	—	—	—	—
Sweden :								
Steamers	2	2,050	1	1,450	1	850	—	—
Motor vessels....	3	11,300	—	—	2	13,260	1	130
America :								
Steamers	18	50,756	10	42,350	10	48,400	8	39,200
Motor vessels....	7	18,200	1	1,000	—	—	2	1,498

during the quarters ending March 31, 1923, December 31, 1922, September 30, 1922, and June 30, 1922, respectively. The figures for the United Kingdom show a distinct increase in motorship tonnage for each of the quarters under review. Thus, for the quarter ended June 30, 1922, there were five motorships, with a total gross tonnage of 810 tons, commenced, whilst for the quarter just ended the number has risen to ten, with a total gross tonnage of 48,024 tons. At the same time, the number of steamships has increased from twelve, with a total gross tonnage of 37,177 tons, to 110, with a total gross tonnage of 303,949, an increase over the period of 366,772 tons for steam vessels and of 47,214 tons for motorships. The motorship is clearly in the minority, but at the same time it is gratifying to note that motorship tonnage is on the increase. In France the only figures showing for the past four quarters are nine vessels, with a total gross tonnage of 3,900, commenced during the quarter ended December 31, 1922. Italy has no motorship figures showing at all for the past four quarters, and this is also the case with Japan, whose steamship figures for the last two quarters show a drop from twelve ships, with a total gross tonnage of 12,234 tons, to five, with a total gross tonnage of 5,530. In America steamship tonnage shows a rise from ten ships, with a total gross tonnage of 42,350, to eighteen, with a total gross tonnage of 50,756; the motorship figure for the past quarter is seven vessels, with a total gross tonnage of 18,200. Of the Scandinavian countries, Sweden is the only one with a big show of motorships, and in point of fact her motorship tonnage laid down, as will be seen, exceeds her steam tonnage for the quarter. Table II summarises the total tonnage for all countries on which work was commenced during each of the quarters mentioned, and it also gives figures showing the motorship percentage of the total. From this it will be seen that there is an increase in the percentage over the quarter ended December 31,

TABLE II.—SHOWING PERCENTAGE OF MOTORSHIPS OF TOTAL TONNAGE FOR ALL COUNTRIES.

	QUARTER ENDING :—			
	June 30, 1922.	Sept. 30, 1922.	Dec. 31, 1922.	March 31, 1923.
	Tons.	Tons.	Tons.	Tons.
Steam	133,416	148,297	291,995	443,716
Motor	10,866	36,875	41,870	79,616
Motorship percentage of Total	7·5	20	12·5	15·2

1922, but that the highest percentage for the four quarters was 20 per cent. for the quarter ended September 30, 1922.

Some of the new motorships embody interesting types and novel departures. Although probably not figuring in the totals already quoted, there is, for example, the motor liner ordered from Sir W. G. Armstrong, Whitworth & Co. Ltd., for the Swedish-American Line. This will have a new type of double-acting Burmeister & Wain engine, to be built at Copenhagen and fitted on the ship at Walker. We have yet, however, to see a fast cross-channel ship with internal-combustion engines.

INDICATOR DIAGRAMS AND THEIR USE.

THE taking of indicator diagrams forms one of the routine jobs on vessels propelled by reciprocating engines, and yet beyond a more or less casual examination of the diagrams, with an occasional determination of the indicated horse-power, very little practical use is made of the diagrams after they have been obtained. It has often been asserted that, to the average sea-going engineer, the uses of the indicator diagram are summed up in the measurement of indicated horse-power and the visualisation of the conditions regarding the setting of the valves, but while this may be true in many cases, it is undoubtedly a fact that to many marine engineers these diagrams yield a wealth of knowledge regarding the general conditions which obtain in his engines and further, they enable him to determine with comparative ease whether the efficiency of the machinery in his charge is as high as it might be. There are, however, many further uses to which indicator diagrams can be put, and even the most studious engineer may not be fully aware of the great wealth of information to which these easily obtained diagrams form the key, or perhaps such knowledge is considered so commonplace that it is not put forward in text books or in papers before our technical institutions as frequently as it might be. Hence the paper entitled "Some Deductions from Indicator Diagrams," presented to the Institute of Marine Engineers during the last session, by Mr. Alberto Keens, was particularly welcome.

Apart from the usual uses to which the indicator diagrams are put, such as the measurement of the mean effective pressure on the pistons and thence the determination of the indicated horse-power, Mr. Keens suggests a number of calculations

which can be made easily with a view to examining the behaviour of the steam as it passes from the main stop valve to the condenser. He bases his arguments upon the usual marine type of triple-expansion engine and from a series of actual diagrams, together with a very complete set of particulars of an engine having cylinders 27 in., 44 in. and 73 in. diam., with a piston-stroke of 48 in., he obtains a vast amount of very valuable information regarding the performance of the engine. Space will not permit of a detailed record of the deductions made, but it will be of interest to mention briefly some of them, so that the reader can obtain an idea of the scope of the subject under review. In the first place Mr. Keens discusses the vexed question of the correct sequence of the cranks, and from a study of the actual indicator diagrams he traces the effect of the distribution of the steam from one cylinder to the next in the H.P.–I.P.–L.P. sequence as compared with that in the L.P.–I.P.–H.P. sequence, and his conclusions appear to favour the latter. The next problem dealt with is that of the effect of cylinder clearance, in which he gives the method of determining the clearance volume from the indicator card. Thence he proceeds to consider the effect of clearance upon the mean effective pressure, the ratio of expansion, and the temperature range in the cylinder. In these calculations he employs the usual law for expansion in the cylinder PV = constant. This is, of course, the law which is generally employed in the elementary text-books on this subject, but the question arises as to whether the more accurate law PV^{γ} = constant should not be employed, taking $\gamma = 1{\cdot}12$ for wet steam and $1{\cdot}2$ for superheated steam. The problem of the effect of compression is then dealt with, this being taken in conjunction with that of clearance, and from these the author proceeds to deal with the drop of pressure in passing from one cylinder to the next. This introduces the important problem of "linking in" and also that of the leakage of steam from one side of the piston to the other, although here it should be noticed that the problem of leakage past the slide valves has apparently been overlooked. The determination from the indicator diagram of the weight of steam at various points during its passage through the engine is of great interest, although due to condensation and re-evaporation, exact determination at any point is a matter of difficulty. Finally, the problems of efficiency of engine and of boiler as well as efficiency ratio are considered, and from the figures thus obtained the general performance of the machinery is known.

We have frequently urged the necessity for a fuller consideration of the problems which underlie the design and operation of the reciprocating engine in view of the large proportion of the new ships which are being built and which will be propelled by engines of this type. This paper of Mr. Keens' certainly points a way whereby a fuller knowledge of the performance of these engines can be obtained, and whence, having realised wherein the various losses occur, it should be possible to improve the design accordingly. In the engine dealt with in this paper an efficiency of $15{\cdot}3$ per cent. was obtained as against a theoretical maximum efficiency based upon the Carnot cycle of $25{\cdot}8$ per cent., so that the efficiency ratio is $59{\cdot}3$ per cent. This is a very creditable performance, but even so, it is one which is capable of improvement, and it is only by a complete knowledge of the conditions obtaining in the engine that it will be possible to eliminate the various preventable losses that occur. If this can be done, and if the efficiency of the reciprocating engine can be still further improved, it will still be able to compete successfully with its more efficient but less-tried rivals. It may be mentioned that at a recent meeting of the Institution of Mechanical Engineers a number of contributions to the question of indicators was read in which the entire subject was treated from various points of view, notably from that of the internal-combustion engine.

THE PROPELLING MACHINERY OF PLEASURE STEAMERS.

WITH one or two notable exceptions, vessels of the above type plying on British rivers and estuaries, have generally been propelled by paddle wheels, due no doubt to necessarily restricted draught making the installation of the usual type of vertical triple-expansion engine, such as would be employed with screw propulsion, a matter of considerable difficulty. The paddle steamer has but little else to recommend it; it is not easy to manœuvre, for the paddles are seldom independent of one another, and with the advent of the steam turbine there appeared to be no reason why the twin-screw arrangement should not be adopted, as in the case of that very successful type of vessel, the fast cross-Channel steamer. It is, however, well known that the directly-coupled steam turbine is not an economical prime mover at low powers, and the *Kingfisher*, which represented an attempt to introduce turbine propulsion to pleasure craft, was not, we believe, an unqualified success. The improvement of mechanical reduction gearing, however, has altered considerably the position of the steam turbine as an efficient engine for ship propulsion. One of the greatest sources of loss, viz., excessive propeller speed is eliminated, and it would appear that other things remaining the same, the small impulse turbine, with mechanical reduction gearing, would form a particularly suitable propelling unit for pleasure craft of the type we have under review.

It must not be forgotten, however, that the pleasure boat is frequently called upon to run astern for considerable distances, and the provision of astern turbines is always a source of loss of efficiency, both from the point of view of weight of machinery and economy in use. A great deal has been written about the efficiency of the electric drive, and particular stress is always laid upon the fact that this method allows of the elimination of the astern turbine while giving considerably greater manœuvring power; in fact, the electric motors can develop the same power when running ahead as astern, a simple change in electrical connections being all that is required to reverse the direction of motion. It is quite possible that an electric installation could be designed employing a high-speed turbine, the weight of which would not differ very much from that of a geared-turbine set of equal power, and while objection might perhaps be taken on account of the cost of the installation, such a set would undoubtedly prove economical in running. There remains the internal-combustion engine as a possible alternative to the steam engine, no matter under what circumstances the latter is employed. The Diesel engine was developed considerably during the War, consequent upon its use in the surface propulsion of submersible craft, and it has been suggested that a pleasure boat having twin screws with a Diesel engine of about 1,000 h.p. on each shaft would be capable of doing all that is ever required of this type of craft. The reliability of these engines is now undoubted, and their particular advantage for this class of work is their exceptionally low stand-by charges. There is no boiler to be kept under steam, since the engines are capable of being started almost immediately from cold. It is possible that objection might be raised on account of the noise and smell, but with efficient silencing and ventilation these could be reduced to almost negligible proportions.

The current month will witness the return to service of a large number of our river and estuary passenger pleasure craft from their winter lay-up. Most of these vessels are considerably past their middle age; the majority are propelled by paddle wheels, and for some, this season may prove their last. We feel therefore that it is not without interest to review the situation and to make brief reference to the methods of propulsion now available for such vessels other than the stereotyped paddle wheel, which is apt to be regarded rather as the accepted thing.

BOOKS FOR THE MARINE ENGINEER AND NAVAL ARCHITECT.

Brief Reviews of Recent Publications.

Motor Boats. By Major F. Strickland, M.I.E.E., M.I.M.E. Pitman's Common Commodities and Industries Series. London: Sir Isaac Pitman & Sons Ltd., Parker Street, Kingsway, W.C.2. 7¼ in. × 4¾ in. × ⅜ in. Price 3s. net.

It is, we must confess, somewhat surprising on opening a book entitled "Motor Boats" to find that the frontispiece is a 10,000 ton motorship. However, as the author states, "for the purposes of this book, all ships propelled by 'I.C.' engines will be considered as motor boats, from the light rowing boat fitted with a 2 horse-power motor, to the large ship carrying 10,000 tons." He deals in the chapters which follow, however, mainly with different aspects of the motor *boat* question, and both his method of treatment and the text matter is quite good. There is a chapter dealing with the historical development of motor boats, a feature which we have previously emphasised as useful in all types of technical books, a chapter devoted to the skimming action of high-speed motor boats and a useful chapter dealing with examples of motor boats in practice.

The book is not a highly technical work, and the author definitely states this in his preface, but we feel that it will appeal to those interested in any way in motor boats, their working and construction.

Line Charts for Engineers. By W. N. Rose, B.Sc. London: Chapman & Hall Ltd., 11, Henrietta Street, W.C.2. 8¾ in. × 5½ in. × ¾ in., 95 pp. Price 6s. net.

Any addition to the already well proven "Directly-Useful Technical Series" is always a welcome one. The author of the latest addition is well known by his work in the same series on "Mathematics for Engineers." In his new work on line charts the author, almost as the natural corrolary of his previous book, shows how greatly the work involved in routine mathematics in practice can be eliminated by carefully chosen and constructed line charts or nomograms. These useful calculating devices require to be brought as much as possible to the notice of practising engineers, and Mr. Rose in his book does so in a manner which cannot fail to appeal.

As far as an exposition of principle is concerned, the book probably contains more information than any similar book of the same price. Moreover, the practical examples which have been selected by the author to illustrate the various types of chart are not only well chosen but quite directly useful.

There is, however, one aspect of Nomography which is not treated in the work and which is intensely useful, practically. This aspect is the automatic method of converting cartesian diagrams into alignment chart form, independent of the particular formula which may be the basis of the cartesian relation. For instance, one very important example which is useful in all branches of engineering is the alignment chart conversion of an ordinary Mollier diagram or steam chart. This omission, however, should not detract from the immediate value of Mr. Rose's work, which we can confidently recommend to our readers.

Sell's Directory of Registered Telegraphic Addresses. Published by Sell's Limited, 168-7, Fleet Street, E.C.4. 7 in. × 10½ in. × 4 in. Price, including supplement, 45s. net.

This compendium should be invaluable for office use. It is divided into three sections. Section 1 gives a list of firms in London and the provinces with their telephone numbers and registered telegraphic addresses. Section 2 gives an alphabetical list of telegraphic addresses. Section 3 is a classified trades' list and there is also a small section which lists information too late for classification.

Industrial Electric Heating. By J W. Beauchamp, M.I.E.E. Pitman's Technical Primers. London: Sir Isaac Pitman & Sons Limited, Parker Street, Kingsway, W.C.2. 4¼ in. × 6½ in. × ⅜ in. Price 2s. 6d. net.

This is a useful little book. The author says in the foreword that there is a large and growing number of useful purposes in industry to which electric heating can be applied, but that information concerning it is rather scattered throughout technical journals. The object of this book, he states, is to bring this information together for the benefit of the engineer and the student. It is informative and well illustrated. Electric furnaces, applications to welding, electric tools and applications to cooking are dealt with, and a bibliography is appended.

Die Bagger und die Baggereihilfsgeräte. Ihre Berechnung und ihr Bau. Vol. I. By M. Paulmann and R. Blaum. Second enlarged edition. Berlin: Julius Springer, Linkstrasse 23-24, W. 9. 8 in. × 10¾ in. × ⅞ in. Numerous plates, diagrams and photographs. Price 20s. net.

Germany, by reason of the geological characteristics of her principal North Sea exits to the sea, and the continual battle which it is necessary for her to wage against sand in the approaches to her most important ports, has been obliged, in the course of her career as a maritime nation, to study thoroughly the problem of maintaining efficient navigable waterways. This really boils down to a study of the design of dredging plant, and the book we have under review gives one a very good idea of German practice in this important branch of engineering, in which the Civil Engineer works hand-in-hand with the Naval Architect. Dredgers and dredging machinery, their design and construction, forms the subject matter of the work, the first volume of which deals with the marine dredger and its necessary fittings. The authors have divided their work into four main sections. Section I is descriptive of dredgers generally, and includes the grab dredge, self-propelling and non-self-propelling, the bow-well and stern-well bucket or ladder dredge, one example of this type being a very ingenious "hinged" dredge composed of two hulls, one containing the machinery and the other the ladder and fittings. The material dredged up is discharged down a shoot on to the machinery hull, whence, by means of a centrifugal pump, it can be discharged into hoppers or on to dumps at will. The section includes also combination bucket and suction dredges, true suction dredgers, the Frühling dredge, the reclamation dredge, the longshoot dredge, the hopper barge, the self-propelling hopper, and the rock cutter. Sections II and III deal with the calculation work necessary in the design of dredgers and their machinery. Section IV covers the actual constructional details of dredger work, including buckets, pumps, &c. There is a very full bibliography appended, and some excellent folding plates of the types of dredgers discussed in the text. The book is an exceptionally complete one, and it is a work which could be studied with advantage by engineers and shipbuilders connected in any way with harbour work.

TRADE PUBLICATIONS.

The H.R. Gear Company Limited, Ulster Chambers, Regent St., W., have sent us a booklet descriptive of the H.R. Reduction Gear and its varied applications.

The Chesterfield Tube Company Limited, Chesterfield, have sent us a leaflet dealing with seamless steel products, both air pipes and cylinders, for gases.

REDUCTION GEARING FOR STEAM TURBINES.

By CAPT. CHAS. W. REED, B.Sc.

(*Concluded from page* 86.)

Despite the care and accuracy of tooth-generating, certain troubles have occurred in marine gearing. The first of these that we will deal with is pitting of the teeth. This may be due to several causes, such as faulty lubrication, inaccuracies of cutting, &c., but as the erosion usually originates just below the pitch line, it appears due to the addendum of one tooth sliding over a localised portion of the dedendum of the other.

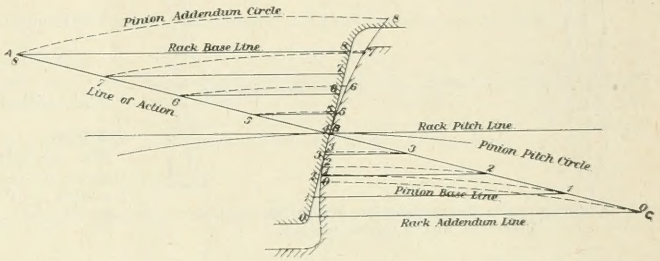
It has been noted that the pitting usually occurs on the pinion rather than on the wheel, which is doubtless due to the fact that the dedendum of the pinion in involute gears makes tooth contact with the wheel during the arc of approach, and it is well-known that there is greater friction during the arc of approach than during the arc of recess.

It is suggested that, to overcome this pitting or erosion, if the teeth were designed with the root-circle coinciding with the pitch-circle, and thus eliminating the dedendum, there could be no contact during the arc of approach, and consequently less friction. Also, if the teeth were designed to increase the rolling and decrease the sliding contact this would greatly help, as in Bostock and Bramley's "Enveloping" teeth.

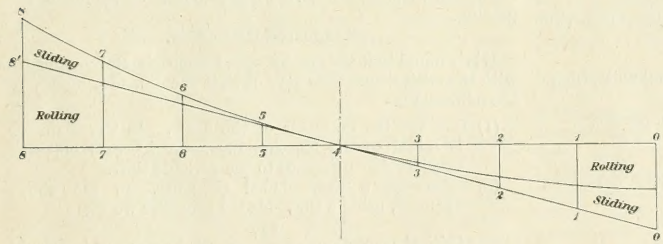
In order to determine this latter, a diagram, showing a comparison between sliding and rolling contact during the tooth-engagement, can be made as follows :—

Referring to the sketches, first draw the profile of the pinion tooth and wheel tooth (in this case, drawn as a rack) at contact on the pitch point. The line of action is represented by ABC. Tooth contact will take place all along this line, which we will divide into eight parts, numbering the positions of contact on the teeth with similar numbers. Then, at point 4 (pitch point) on line of contact, both sliding and rolling will be nil, so at any other point 8 (say), the amount of sliding which has taken place is equal to length 4 to 8 on pinion profile, minus the length 4 to 8 on the rack profile, because point 8 on rack and point 8 on pinion meet at point 8 on the line of action. In the lower diagram the vertical distance 8 to 8 is equal to length of pinion profile 4 to 8, and the distance 8 to 8 is equal to length of rack profile 4 to 8 and so on.

The diagram thus shows a comparison between sliding and rolling at each side of the pitch point for the particular design of teeth.



Sketch showing Contact of Pinion and Wheel Tooth.



Diagrammatic Representation of Contact.

Torsional Vibrations.

More important than this pitting, which is not often very serious, is the occasional stripping of teeth, which, it is believed, is due chiefly—omitting failures due to faulty design and material—to the vessel running at a speed when the synchronization of torsional vibrations of the system occurs. Torsional vibrations, as distinct from elastic or transverse vibrations, are set up in a shaft connected to a propeller and engine (or gear-wheel) due to variations in the torque. It has been demonstrated, both by theory and experiment, by Dr. Otto Schlick (Inst.N.Arch, 1911), Dr. Gümbel (Inst.N.Arch., 1912) E. N. Janson (Am. Soc. Nav. Eng., 1913), and R. J. Walker, (N.E. Coast Inst.E.S., 1919, Inst.Nav.Arch, 1921), that each propeller blade experiences the greatest resistance when in its upper vertical position, and thus produces one fluctuation or vibration per revolution. As the speed increases these impulses delivered through the shaft approach and are finally exactly in phase with the natural torsional period (or "tone") of the system, and a greatly aggravated form of vibration ensues. The speed at which this occurs is called the critical speed of synchronous torsional vibrations, and does not depend on the bearings, nor on the power transmitted. The natural period of vibration of the system may be calculated from various formulæ, which will now be discussed, and if we then find the number of revolutions corresponding to the period of forced vibrations set up by the blades, we can find the revolutions at which synchronization occurs.

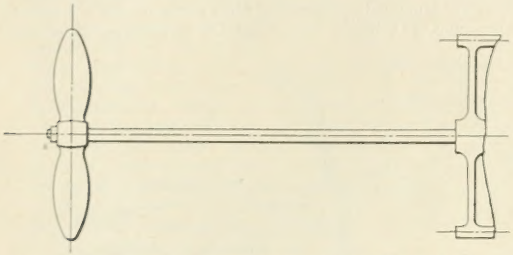
If N = No. of natural vibrations per minute.
F = No. of forced vibrations per minute.
= Revs. per minute × No. of blades.
When F = N, we have synchronization.

$$\text{Therefore critical speed} = \frac{N}{\text{No. of blades.}}$$

The question of determining the natural period of the system is very complex, but without being unduly categorical we suggest that the following may be taken as the general principles.

We have under consideration a system consisting essentially of a long flexible shaft connecting a heavy propeller to a heavy gear-wheel, which is driven by pinions coupled to high-

speed turbines. Now, due to torsional variations, vibrations will be set up in the system, having, if the gear-wheel is not restrained, a node or spot of no vibration somewhere near the centre of length of shaft, the exact position depending upon the moments of inertia of the gear-wheel and propeller. This will be the natural condition of the system when running at a certain speed, which we desire to find. However, the vibrations which would occur in the gear-wheel at this critical speed are damped or restrained by the pinion teeth in engagement with it, as the pinions are coupled to high-speed turbines consequently having great inertia, and thus we get a forced node at the gearing. It follows therefore, that, for the purpose of calculation, we may take the shaft as being fixed at the



Diagrammatic Representation of the Mechanical System under Consideration.

gear-wheel boss and twisted by the propeller. We are now in a position to find the period of natural vibration of the system from a consideration of periodic rotational motion.

We have the elementary formula :—

The Periodic Time, P, in seconds, for a complete oscillation

$$P = 2\pi \sqrt{\frac{\text{Angular Displacement}}{\text{Angular Acceleration}}} \quad \ldots\ldots\ldots\ldots\ldots\ldots(1)$$

Using the following symbols—

I_p = Moment of inertia of the vibrating mass (the propeller) in (inches)^4 units.

$I_m = \frac{I_p}{g}$ = Moment of inertia in gravitational units, where

g = Acceleration due to gravity = 386·4 in. per sec.
G = Modulus of rigidity of the shaft.
d = Diameter of the shaft.
l = Length of shafting, from aft end of gear-wheel boss to centre line of propeller.

All in inches and pounds.

$$\text{The angular acceleration} = \frac{\text{Twisting moment}}{\text{Moment of inertia}} = \frac{T}{I_m} \quad \ldots(2)$$

where θ_r = Twist in radians

$$\text{Twisting moment} = \frac{\theta_r \times G \times I_p}{l}$$

$$= \frac{\theta_r \times G \times \pi d^4}{l \times 32}$$

For one radian of displacement, *i.e.*, $\theta = 1$,

$$\text{Twisting moment} = \frac{\pi}{32} \times \frac{d^4 \times G}{l}$$

$$\text{Therefore from } \ldots\ldots\ldots (2) \text{ angular acceleration} = \frac{T}{I_m}$$

$$= \frac{\frac{\pi}{32} \times \frac{d^4 \times G}{l}}{I_m}$$

Now, from ..... (1)

$$\text{Periodic time} = 2\pi \sqrt{\frac{\frac{1 \text{ Radian}}{\frac{\pi}{32} \times \frac{d^4 \times G}{l}}}{I}}$$

$$= 2\pi \sqrt{\frac{I_m \times 32 \times l}{\pi d^4 \times G}}$$

Substituting for I

$$P = 2\pi \sqrt{\frac{I_p \times 32 \times l}{\pi d^4 \times G \times 386 \cdot 4}}$$

Taking G, for steel = $11{\cdot}75 \times 10^6$ lbs./per sq. inch

$$P = 2\pi \sqrt{\frac{I_p \times l}{d^4 \times 10^6 \times 445 \cdot 8}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(3)$$

The frequency of vibrations per minute = f

$$f = \frac{60}{P}.$$

Then critical revolutions per minute = $\frac{f}{4}$ for a four-bladed propeller.

For example, find the critical speed of a geared-turbine steamer having a four-bladed propeller of 19 ft. diam., and $I = 61{\cdot}5 \times 10^6$ lbs./ins^2. Shaft = 17 in. diam. and 38 ft. long.

$$P = 2\pi \sqrt{\frac{61 \cdot 5 \times 10^6 \times 38 \times 12}{(17)^4 \times 10^6 \times 445 \cdot 8}}$$

$$= 2\pi \sqrt{\cdot 000754}$$

$$= \cdot 1723 \text{ seconds.}$$

$$f = \frac{60}{\cdot 1723} = 347 \text{ per minute.}$$

$$\text{Critical Speed} = \frac{347}{4} = 86{\cdot}75 \text{ r.p.m., four a four-bladed screw.}$$

So that we see, if the vessel is normally running in the region of 85 r.p.m., she is very likely to develop trouble in the gearing.

Suggested Remedies.

If it is found that the normal speed approximately coincides with the critical speed we may obviate the difficulty by any of the following :—

(1) Increase the diameter of the shaft. As the diameter is raised to the fourth power, any increase makes a considerable difference to the periodic time.

(2) Substitute a three-bladed propeller. In the above example, using a three-bladed propeller we get—

$$\text{Critical revolutions} = \frac{347}{3} = 115{\cdot}7 \text{ r.p.m., which would}$$

allow a good margin over the running speed.

(3) A method suggested, and known as the "Flexible Pinion" or "Nodal" drive, consists of the arrangement shown in the accompanying sketch.

The idea seems to lie in the suggestion that, having found the critical revolutions of the main shafting, then, in order that the teeth of the gearing will not separate when vibration takes place, the first-reduction pinion should have the same period of vibration as the propeller shaft. This object is attained, it is claimed, by driving through a long flexible shaft, which passes through the hollow pinion and is coupled by a flange at the aft end only. Other things being equal, the length of the pinion shaft varies directly as the length of the propeller shaft. The length required is calculated as in the following example :—

Find the moment of inertia of L.P. rotor = 80,800 lbs./ins.^{2}

The inertia of H.P. rotor is made equal to this by the addition of a flywheel.

Then period of vibration of propeller shaft = ·43 (say).

This period of vibration also equals $1{\cdot}81 \sqrt{\frac{I_R \times l.}{I_S \times G.}}$

where I_R = moment of inertia of rotor.
I_S = moment of inertia of shaft.
l = Length of shaft required.
G = Modulus of rigidity; for nickel steel, say, 13×10^6 lbs./ins^2.

Shaft to be $3\frac{3}{4}$ in. diam. (say).
Then equating

$$\cdot 43 = 1{\cdot}81 \sqrt{\frac{80800 \times l}{13 \times 10^6 \times 19{\cdot}41.}}$$

Therefore length of shaft required l = 176·5 in. = 14 ft. $8\frac{1}{2}$ in.

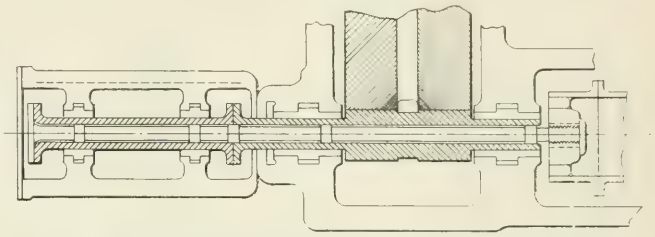
The apparent disadvantages of this system are :—

(*a*) Long pinion shafts.

(*b*) Hollow shaft, giving a larger deflection; which, as surface speed of journals are already near the maximum, cannot be compensated by an increase of diameter.

(*c*) Liability of the hollow-shaft partially to grip the inner shaft, if any bending or vibration occurs, thus entirely neutralising the whole idea.

(*d*) Only the length of the inner shaft is calculated, as the outer shaft is taken to be so stiff as to have no vibration, which is absurd.



Sketch showing Arrangement of Flexible Drive and Nodal Pinion.

(4) If it is difficult in anyway to provide a large margin over the critical speed by any of the above means, it is advisable to fit a large and heavy gear-wheel, with a comparatively large moment of inertia to damp out any vibrations. This has been found in actual practice to be very beneficial.

SEA TRIALS AND BLOCK COEFFICIENTS.

The paper by Mr. G. S. Baker on "Measured Mile Trials and other Ship Propulsion Data," read before the North-East Coast Institution of Engineers and Shipbuilders in March, dealt exhaustively with the discrepancies between the actual and estimated powers required for propulsion, but neither in the paper nor in the subsequent discussion was any reference made to an important point in connection with measured mile trials, to which, in our opinion, due consideration has not been given. Mr. Baker, referring to such trials, says, "The object in most cases is to demonstrate the steaming qualities of the ship and the maximum power of the ship's engines." Precisely, but what is the effect on design of this popular system? The shipowner, having in his mind a certain mean sea speed for his proposed new vessel, specifies a definite "speed on trial," which may be half a knot or even one knot greater. If this speed is not attained on trial, there is frequently a financial penalty involved, and in any case there will be a certain amount of what would be vulgarly termed "back-chat" between builder and owner, with possible prejudice against the builder when the owner contemplates placing a further order. For these reasons, the vessel must be designed for that trial trip speed, and the vessel's form, propeller characteristics, bore and stroke of engines, diameter, heating surface and grate area of boilers are all determined with a view to ensuring that the specified speed on the measured mile shall be obtained. In other words, the vessel is designed for a condition which has at the outside a duration of, say, two hours, and for a speed which is not even required to be maintained continuously during that short period. If it happens that the vessel, as designed for these trial conditions, possesses also precisely that form and power and propeller most suitable for normal sea conditions, this must obviously be regarded more as a happy coincidence than as a carefully pre-arranged result.

Probably the most serious argument against measured mile trials is to be found in the undesirable and artificial restriction placed on form in general and block coefficient in particular by the specification of a trial trip speed. No alternative is left to the designer but to determine these characteristics with a view to this required speed. Different designers have different methods of arriving at the most suitable block coefficient, but it is generally agreed that it is governed by the speed and length of the ship. A good working formula, as used by many designers, is that derived by Mr. F. H. Alexander, M.Sc., namely, block coefficient $= 1{\cdot}06 - \frac{V}{2\sqrt{L}}$, where V is the maximum speed required and L the length of ship. Clearly the determination of block coefficient with a view to a mean sea speed, rather than a trial trip speed is preferable, since in the majority of cases it would mean a slightly fuller coefficient, which should, in general, prove more favourable to the shipowner. It is not a sound argument to claim that the specification of a trial speed will ensure good performance in rough weather. An economic design should always be based on average rather than extreme conditions.

It is unquestionable that the correct choice of block coefficient is a matter of extreme importance, since it is known that the profit-earning capacity of a vessel is highly sensitive to small changes of coefficient. Almost every factor governing the profit-earning capacity varies with block coefficient as, for example, displacement, deadweight, net tonnage, machinery weight and cost, fuel consumption and cost, hull weight and cost, speed, &c.

Obviously, the shipowner, rather than the shipbuilder, is best in a position to gauge, by careful analysis of previous vessels, the most suitable coefficient for a proposed vessel. It would be sound policy for the shipowner who wishes to avoid the ill-effects of the present system, to specify in each case a definite block coefficient and, if possible, to arrange for long-course trials in place of the customary measured mile trials.

THE ITALIAN PASSENGER LINER "CONTE VERDE."

Details of the Propelling Machinery and Boilers.

The t.s.s. *Conte Verde*, a sister ship to the *Conte Rosso*, delivered to her owners about a year ago, underwent her steam trials successfully last month. She has been designed and constructed by William Beardmore & Co. at their Naval Construction Works, Dalmuir, for the Lloyd Sabaudo Company, Genoa, for their passenger and cargo service between Italy and South or North America. After completing her trials in the Clyde, the vessel left for Genoa which she reached without mishap on Monday, April 16, having made an average speed of 19 knots from the Clyde. She encountered heavy weather in the Gulf of Lyons, and it is interesting to note that the speed of 19 knots mentioned was maintained easily in spite of the adverse weather conditions; the contract service speed specified for the ship was 18½ knots. The general particulars of the vessel are as follows :—

Length, overall	593 ft.
,, on l.w.l.	580 ft.
,, b.p.	570 ft.
Breadth, moulded	74 ft.
Depth, moulded, to shelter deck	40 ft.
Load draught	26 ft.
Gross tonnage	18,500 tons
Speed on service	18½ knots
Number of passengers and crew	2,830

The vessel is of the shelter-deck type (without tonnage opening) and she has been built under special survey to the highest class of Lloyd's, British Corporation, and the Registro Navale Italiano. There is an interesting point of difference in the construction of this vessel from that of the *Conte Rosso*, her sister ship. The *Conte Rosso* has expansion joints in her superstructures, but in the *Conte Verde* these have been dispensed with, the superstructures have been strengthened, and the boat deck has been made the strength deck. The vessel has a straight stem, well raked forward, an elliptical stern, two funnels with cowl tops, and two pole masts.

The Propelling Machinery.

The main propelling machinery consists of two sets of compound steam turbines of the Parsons impulse and reaction type, each set consisting of one high-pressure turbine and one low-pressure turbine, and driving, through double-reduction double-helical gearing, a common gear wheel mounted on the propeller shaft. Astern turbines are incorporated in each of the low-pressure turbine casings. The turbines are arranged to operate at 185 lb. steam pressure and 200 deg. Fahr. superheat at the turbine stop valves. The ahead and astern portions of the turbines, where subject to superheated steam, are of cast steel, and the blades are of special phosphor-bronze, in order to counteract any difficulties which might be experienced from the action of high temperatures on these parts. The designed speed of the high-pressure turbines is 2,950 r.p.m., and that of the low-pressure turbines 1,686 r.p.m., and it is interesting to note that these speeds were exceeded by 20 per cent. during steam trials in the shops. Each low-pressure turbine exhausts into an underslung condenser, an arrangement frequently adopted now in large units, which has the advantage of reducing the passageway for exhaust steam to a minimum. Propeller racing is guarded against by the fitting of Cockburn-Aspinall governors, which operate also to shut off steam in the event of a failure of the forced lubrication system or a drop in oil pressure below a predetermined limit. The turbine manœuvring platform, which we illustrate opposite, is at main turbine level, and not sunk to a flat slightly above the tank top, as has been the practice in many recent vessels. The new arrangement enables the engineering staff to have complete survey of the machinery and immediate access to running parts without leaving the control valves.

Auxiliary Machinery.

The condensing plant is of standard type, comprising Weir dual-type air pumps and Paul's centrifugal circulating pumps, the latter driven by high-speed enclosed forced lubrication engines. The forced lubrication system is of the pressure-gravity type, and the pumping outfit for this consists of four Weir pumps, one of these pumps being used for stand-by purposes. There is also a De Laval centrifugal separator fitted for oil-cleaning purposes. A workshop is fitted for the purpose of enabling the ship to be independent of the shore for small repairs. The equipment of the workshop includes a motor-driven lathe, shaping machine, drilling machine, emery wheel, and grindtone, &c., and the arrangement of this has been carried out to secure maximum efficiency of operation.

Boiler Rooms.

Steam is supplied by six double-ended and two single-ended boilers of cylindrical type, manufactured by the engine builders at their Dalmuir works. The single-ended boilers have a diameter of 16 ft. 3 in. and a length of 11 ft. 3 in., and the double-ended boilers are 22 ft. long, with the same diameter. The single-ended boilers have four furnaces each, and the double-ended, eight furnaces. All the boilers are fitted with Howden's forced draught and Robinson smoke-tube superheaters, and they generate steam at 200 lb. per sq. in. pressure and 220 deg. Fahr. superheat. The tubes are 3 in. in diameter and there is a total heating surface of 42,154 sq. ft. Primarily, the boilers are designed to burn oil on the Wallsend-Howden system, but they are capable of rapid conversion to coal-firing in case of necessity. Provision for changing the furnace fronts and for stowage of coal in lieu of oil has been made so that either type of fuel can be used. There are two separate boiler rooms, each having three double-ended and one single-ended boiler abreast, and a separate funnel is provided for each group. The oil-burning plant comprises duplicate units in each boiler room, one of which is capable of supplying all the boilers in its own compartment. The recording and measuring apparatus includes pneumercators, distance thermometers, recording gauges, CO_2 recorders, torsionmeters, tachometers, &c., and is very complete.

Electric Generating Arrangements.

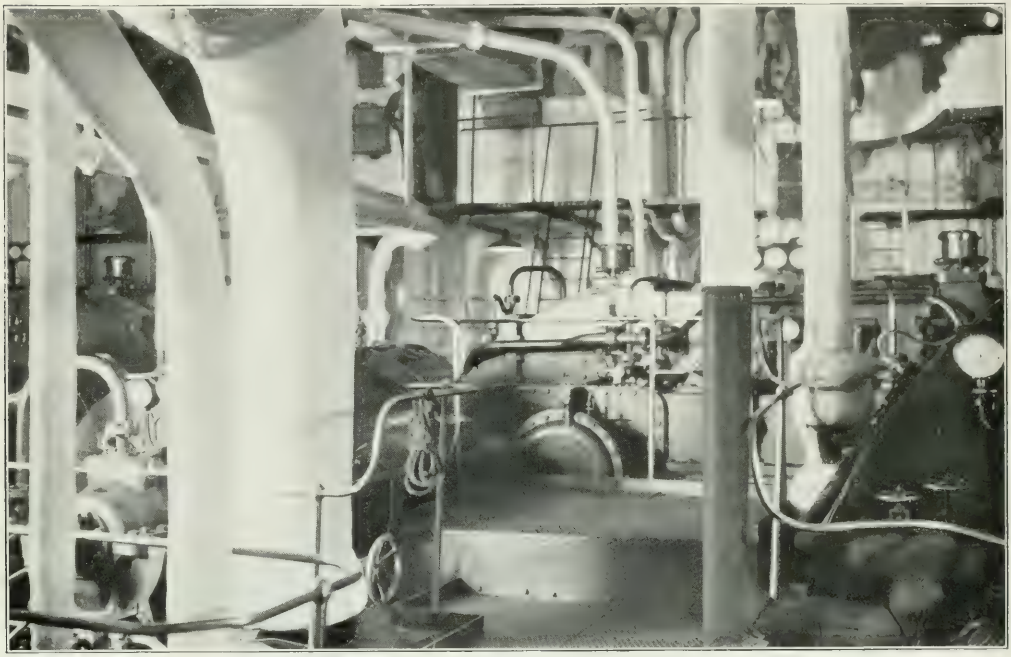
The electric light and power arrangements are as follows :

(*a*) *Main generators.*—These consist of three 150 kw. 110-volt geared turbo-generators.

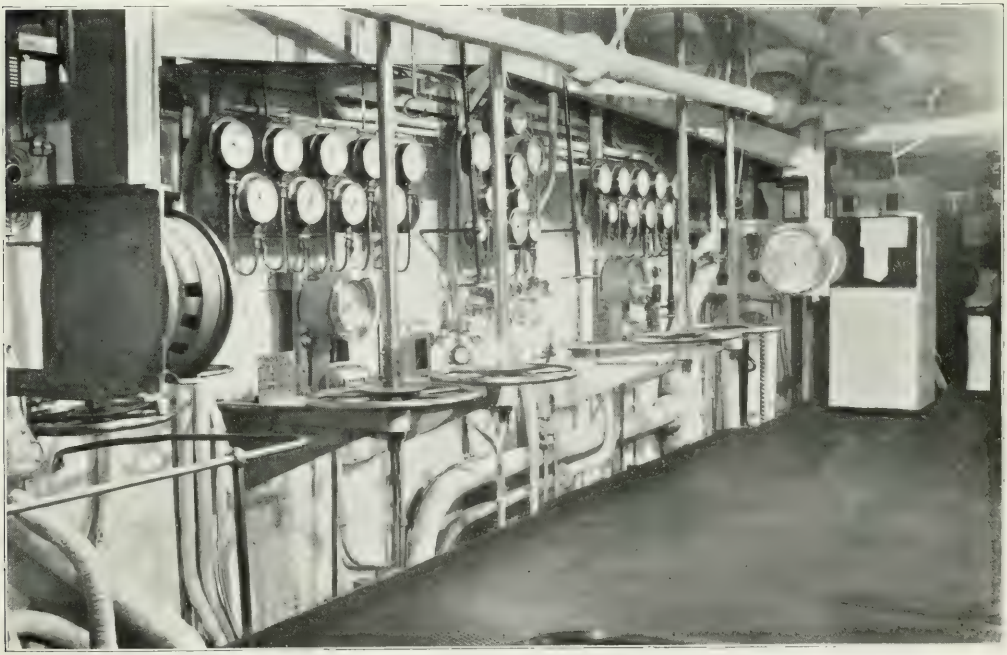
(*b*) *Emergency generator.* This is a 35-kw. 110-volt oil engine-driven generator.

The scheme of distribution is ring-main, feeding automatic circuit breakers, remote controlled. The *Conte Verde* is the second merchant vessel to which the Admiralty practice of fitting a ring-main has been applied, the first being the sister vessel *Conte Rosso*. It consists of one positive and one negative cable, each having between frames 62 and 103 a cross-section of 2 sq. in., and between frames 103 and 150 a cross-section of 1 sq. in. The cross connecting cable at frame 103 has a section of 2 sq. in. All the ring-main cables are run on the main-deck level.

THE ENGINE ROOM OF NEW LLOYD SABAUDO LINER "CONTE VERDE."



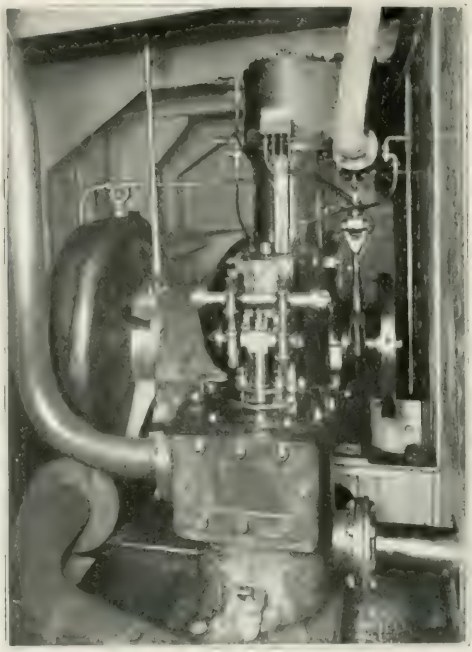
View of Engine Room, looking Aft, showing Port Turbines (H.P. and L.P.) and Gear Case.



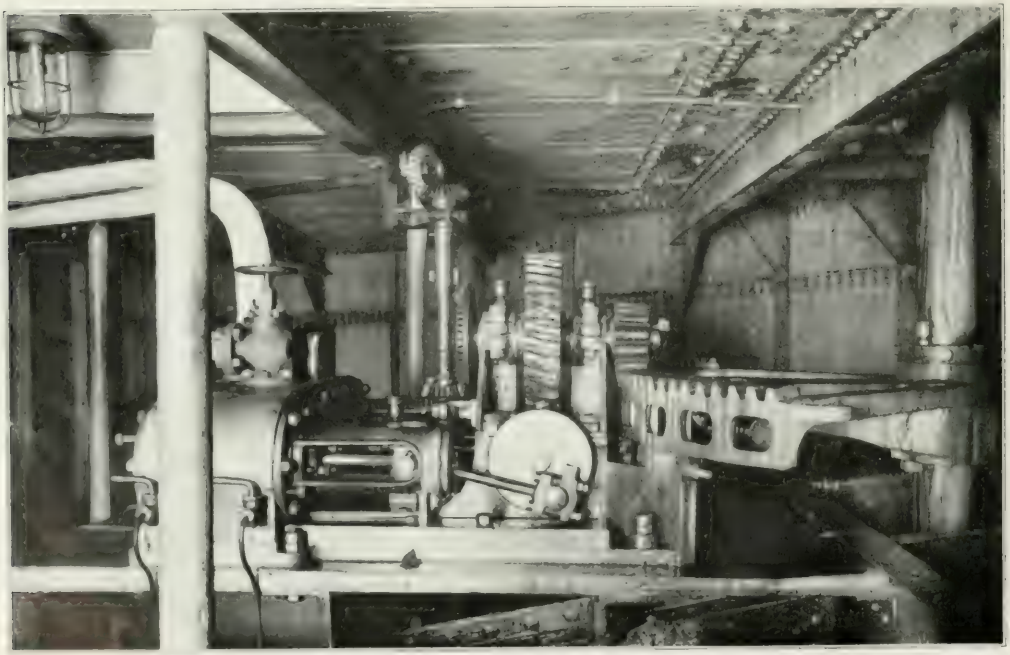
View of the Control Platform, showing Manœuvring Controls, Telegraphs, Revolution Counters, &c.

THE THAMES FERRIES "SQUIRES" AND "GORDON."

The "Gordon." View showing the New Type of Stern.

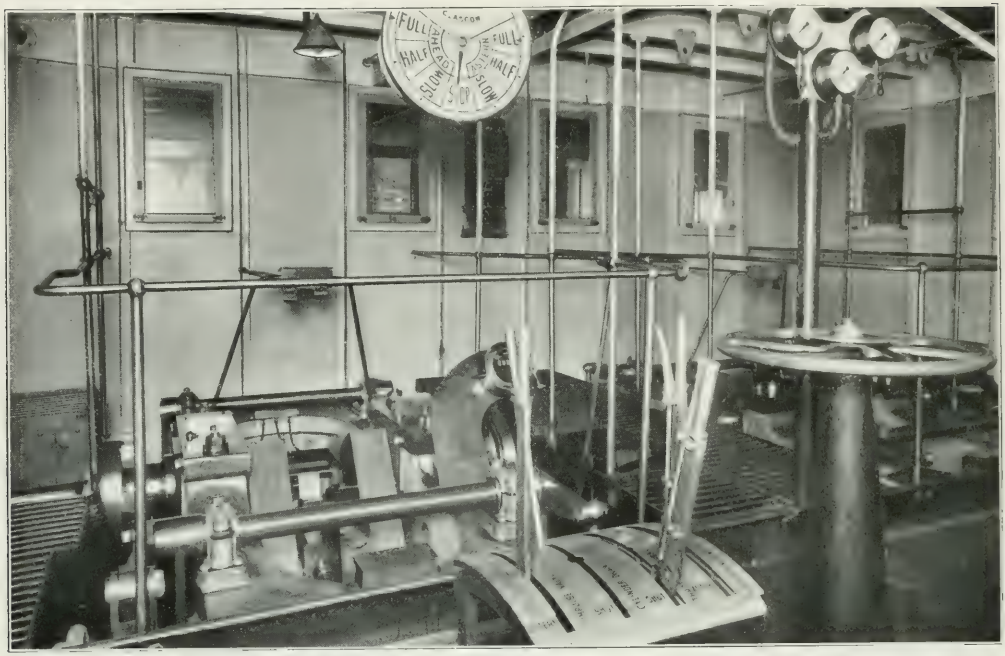


The "Squires." Air and Circulating Pumps driven off Common Engine.

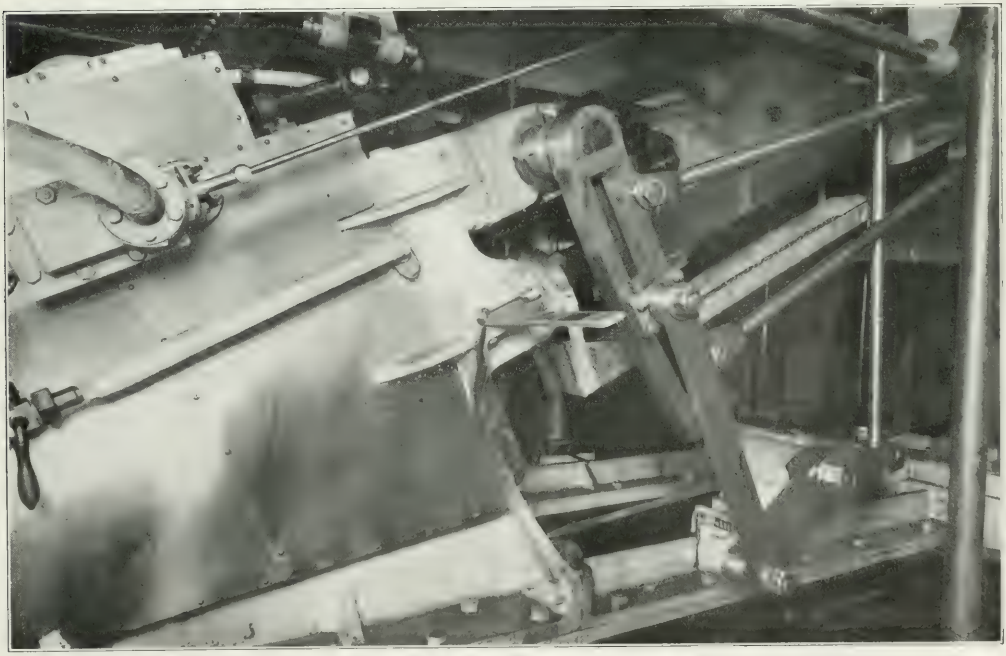


The Direct-Acting Steering Gear of the "Squires," showing Control Rods.

PLYING BETWEEN WOOLWICH AND NORTH WOOLWICH. (See page 191.)

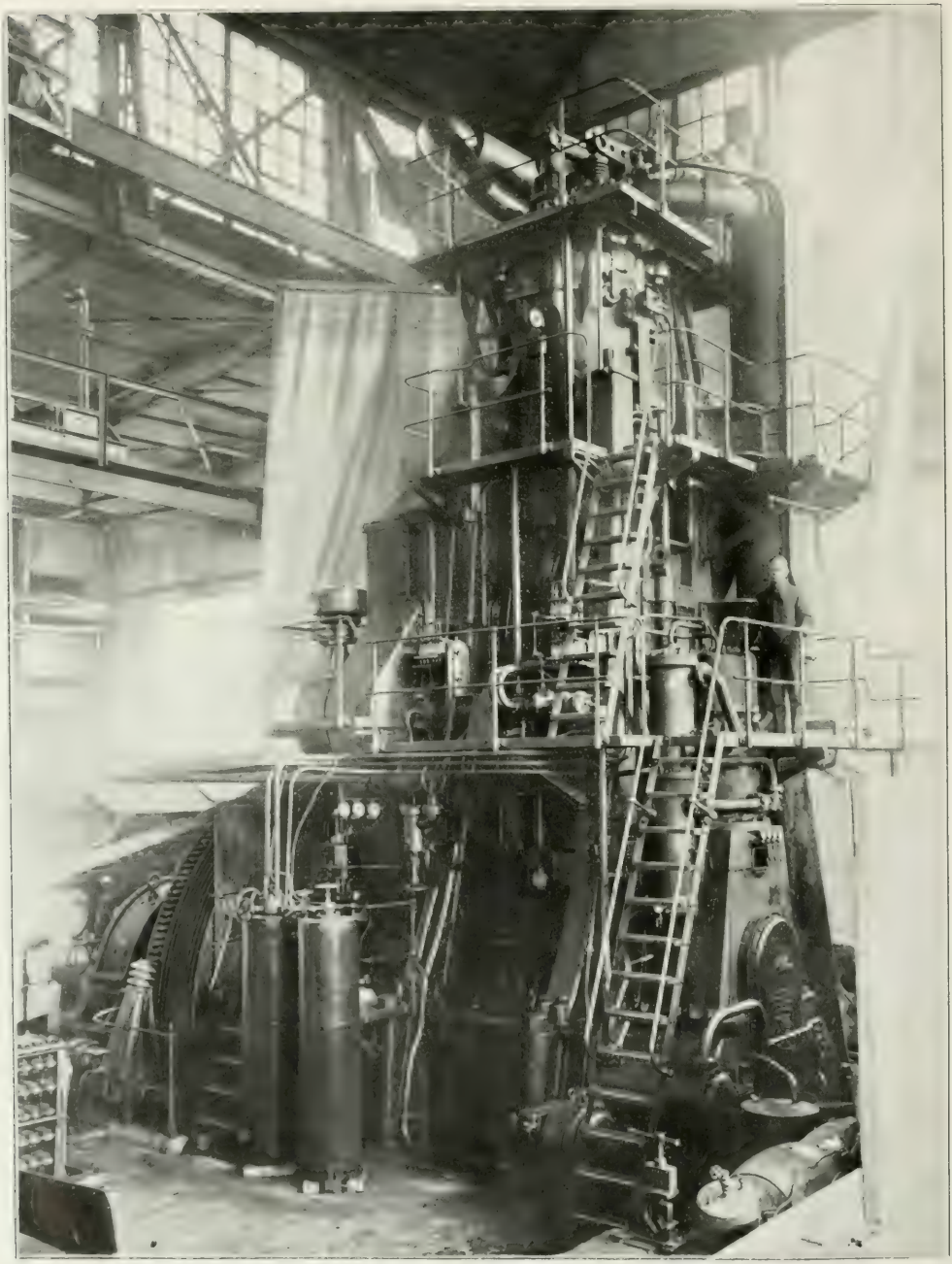


The Thames Ferry "Squires." View of Control Platform, showing Manœuvring Levers, &c.



One of the Cylinders of the "Squires'" Main Propelling Engines, showing Reversing Gear.

THE NEW TYPE OF DIESEL ENGINE FOR THE SWEDISH-AMERICAN LINER.



Experimental Double-Acting Marine Diesel Engine at Burmeister & Wain's Works, Copenhagen. (See page 192.)

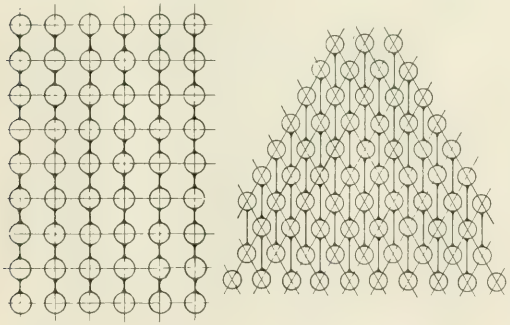
THE EFFECT OF CONDENSED STEAM ON THE EFFICIENCY OF A CONDENSER.

The Ginabat Condenser with Improved Tube Arrangement.

It has been generally recognised that the efficiency of a steam turbine is dependent upon that of the condenser, and that the higher the vacuum produced by the condenser, the better are the working conditions in the turbine. Two main factors have so far prevented marine engineers from producing in the condenser a vacuum corresponding to the barometric height; these being, first, the presence of air accompanying the steam, and second, the streaming of condensed water from the upper to the lower tubes of the condenser. Serious efforts have been made in order to eliminate air from the water in the boilers, and good results have been obtained from these. But no arrangement has so far been devised which has proved effective in reducing the loss of efficiency caused by the pouring of condensed water on the condensing surfaces.

There are two methods which may be considered with a view to avoiding the inconvenience resulting from the existence of drops of water on the tubes of the condenser; the first one consists of inclining the whole of the tubes of the condensers in order to facilitate the flowing of water down the tubes, but such a proceeding involves many complications in the arrangement of the condenser plant, particularly in the case of marine condensers, and it seems that no practical results can be obtained by this method.

Another solution had therefore to be found, and it was considered that an improvement might be obtained by a re-arrangement of the tube. This method has been employed with considerable success by an engineer in the French Navy, M. Ginabat.



Figs. 1 and 2.—Showing Two Typical Arrangements of Tubes in Marine Condensers.

From Fig. 1 and Fig. 2, which represent the two arrangements of tubes generally employed in marine condensers, it is seen that the water condensed on the surface of each tube forms a drop at the lower part of the tube, which falls on to the upper part of the tube situated just below. The speed of this falling water is thus considerably reduced, and it is obvious that the drop, after falling on the upper part of a tube, is divided into two parts, which flow on each side of the tube, thus covering the whole surface of each tube. Supposing that the tubes of a condenser are disposed according to the arrangement shown in Fig. 2, and that the condenser is composed of 100 horizontal rows of tubes, which is an average for a big marine condenser, it will be seen that each tube of the lowest row receives on its upper part the water pouring from 49 tubes. Taking into account the surface of the tubes and the average weight of steam condensed per square foot in the condenser, it is easy to calculate the weight of condensed water pouring on each of the tubes of the lowest rank. As an average, it may be thus computed that the weight of condensed water pouring on each of these tubes amounts to about 176 kg. per hour per metre length of tube (130 lb. per hour per foot). It was established that in the case of a tube of 22 mm. external diam. (0·86 in.), the drop of water pouring on the upper part of a tube was reconstituted

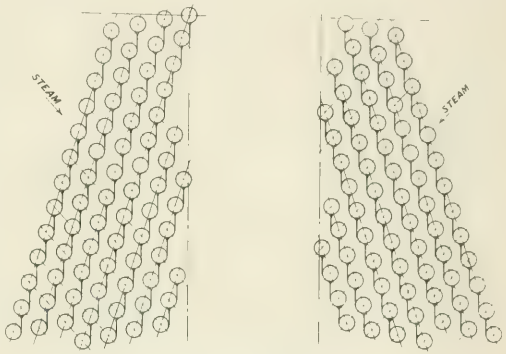


Fig. 3.—Showing Tubes as arranged in the Ginabat Condenser.

on the lower part, after flowing on the two sides of the tube, in about one second. From the above figures, it may be calculated that the weight of water pouring on each of the tubes of the lowest rank is about 66 gms. (2⅓ oz.) per second. It has been further calculated from these figures that the thickness of the film of water which surrounds at any time the tubes of the lowest rank is about 1 mm. (·04 in.).

The approximate calculation of the thickness of the film of water which covers the tubes of the lowest rank in a condenser is of the highest importance, as, owing to the relatively great thickness thus determined, it can easily be proved that the condensed water pouring on the tubes has a prejudicial effect on the condensation of steam. It may be recalled that, according to the experiments made by Mollier, Joule, and Nichol, it was found that the coefficient of transmission of heat was much lower for a tube placed horizontally than for the same tube placed vertically, the difference being caused by the prejudicial influence of the condensed water which, in the second case, flows on the surface of the tube. It will thus be realised that the presence of condensed water on the surface of the tube of a condenser has an insulating effect. Every effort should thus be made with a view to reducing to the lowest possible extent the surface of the tubes in contact with water, and also the duration of such contact.

The arrangement of the condenser tubes as suggested by M. Ginabat, is shown in Fig. 3. According to this design, the drop of condensed water falls on to the tube situated below tangentially, so that only one fourth of the circumference of the tubes is covered with water. Thus, instead

of tubes in the whole of the tubes of the condensers on the horizontal plan, M. Ginabat thought it more convenient to incline, or stagger, the rows of tubes. Thanks to this innovation, the speed of the water pouring on the tubes is maintained all along its course. The condenser of the Ginabat type is constituted by coupling the two sets of tubes shown in Fig. 3, on which the vertical lines represent the flow of the drops of water. It will be seen that the dry part of the surface of each tube is turned towards the flow of the steam, this dry surface being the most efficient for condensing purposes. The air driven by the steam is collected between the two sets of tubes, and it spreads down vertically towards the air-ejector suction situated at the lower part of the condenser, without it being necessary to provide for special space free from any tube. Further, the condensed water is heated by the steam and is exhausted from the condenser at a higher temperature than in condensers of the usual types, which is another advantage of the Ginabat condenser.

TABLE I.—*Comparative Tests of Ginabat Condenser and Condenser of ordinary design with full load on Turbine.*

	Ginabat Condenser.	Ordinary Condenser.
Load (kilowatts)	27,000	27,000
Steam condensed per hour (kilos.)	138,000 (304,000 lbs.)	131,000 (288,000 lbs.)
Temperature of circulating water (inlet)	7° C. (44·6° Fahr.)	6° C. (42·8° Fahr.)
Temperature of circulating water (outlet)	11·5° C. (52·6° Fahr.)	10·3° C. (50·5° Fahr.)
Temperature of steam on entering condenser	23° C. (73·4° Fahr.)	27·2° C. (81° Fahr.)
Vacuum in condenser (barometric height : 760 mm. = 29·9 in.)	739 mm. (29·1 in.)	733 mm. (28·8 in.)
Temperature of condensed water	16·5° C. (61·7° Fahr.)	10·8° C. (51·4° Fahr.)

TABLE II.—*Comparative Tests of Ginabat Condenser and Condenser of ordinary design, with two-thirds load on Turbine.*

	Ginabat Condenser.	Ordinary Condenser.
Load (kilowatts)	20,000	14,000
Weight of condensed water per hour (kilos.)	104,000 (229,000 lbs.)	68,000 (150,000 lbs.)
Temperature of circulating water (inlet)	6° C. (42·8° Fahr.)	6° C. (42·8° Fahr.)
Temperature of circulating water (outlet)	9·5° C. (49·1° Fahr.)	8·4° C. (47·1° Fahr.)
Temperature of steam on entering condenser	21° C. (69·8° Fahr.)	25° C. (77° Fahr.)
Vacuum in condenser (barometer : 760 mm. = 29·9 in.)	741·5 mm. (29·2 in.)	736·4 mm. (28·9 in.)
Temperature of condensed water	17° C. (62·6° Fahr.)	8·5° C. (47·3° Fahr.)

The Société des Condenseurs Delas" which holds the patent of the Ginabat system, has supplied the five condensers which are fitted to the five main generator sets at the central station at Gennevilliers, Paris, belonging to the "Union d'Electricite." Four of these condensers are of the Ginabat type, the other being of the ordinary type. Each of these generators is driven by a turbine which exhausts about 160,000 kg. (352,000 lb.) of steam per hour, the power being 40,000 kw. The results obtained from the Ginabat condensers are shown in the tables opposite, from which the advantages of the Ginabat condenser are apparent. The first table was obtained for nearly similar conditions in both the ordinary condenser and the Ginabat condenser, and it will be seen that, although the weight of steam received by the ordinary condenser is less than the weight of steam received by the Ginabat condenser by 7,000 kg. (15,400 lb.), the vacuum obtained by the latter is higher by 0·24 in. than is the vacuum obtained by the ordinary condenser. It has been calculated that a difference of 0·393 in. of vacuum in the condenser results in a decrease of the consumption of the turbine of 3 per cent., so that it may be said that the economy obtained by the Ginabat design is :—

$$3\,\% \times \frac{0{\cdot}24}{0{\cdot}393} = 1{\cdot}8\,\%.$$

Taking into account the extra weight of steam, one may compute that the economy resulting from the Ginabat system is about 2 per cent.

A further economy results from the higher temperature of the condensed water, which is 5·7° C. (10·2° Fahr.). The heat supplied to the boilers in this way may be estimated as 0·9 per cent. of the total, so that the whole economy of the system appears to be :

$$2 + 0{\cdot}9 = 2{\cdot}9\,\%.$$

The second table shows that, even when the generators are not working at full load, the economy is maintained; the Ginabat condenser, although condensing 36,000 kg. (79,000 lb.) of steam more than the ordinary condenser, gives a vacuum which is better by 0·20 in. to that obtained in the ordinary condenser. On the other hand, the temperature of the condensed water is 8·5° C. (15·3° Fahr.) higher for the Ginabat condenser than for the ordinary condenser.

It may be pointed out that while it should be possible to obtain an improvement of the vacuum in the ordinary condenser by increasing the surface (the results of the Ginabat condenser might be obtained with an ordinary condenser by increasing the surface by 39 per cent.), an increase of the temperature of the condensed water would not be obtained in this way. The increase of temperature is, therefore, the real characteristic of this type of condenser.

The French Navy has shown great interest in the invention of M. Ginabat, and it is reported that Ginabat condensers will be fitted on the cruisers to be built in accordance with the Naval programme. In the meanwhile, the Navy has verified the efficiency of these condensers by making experiments at the works at Indret.

THE DRAINING OF MARINE STEAM TURBINES.

In the operation of steam turbines, it is essential that water should not be allowed to accumulate at the bottom of the casings. Where the turbine is mounted directly over the condenser it is easy to avoid this, but where the condenser is arranged at the side of the turbine or in compound turbines where the steam from the high-pressure unit passes to a separate low-pressure unit before entering the condenser, arrangements must be made so that the water which comes over with the steam or which is formed by condensation in the turbine itself can drain away to the lowest point of the casing and thence to the engine-room. The use of superheated steam greatly reduces the risk of damage due to water in the turbine casings, but even when a high degree of superheat is employed, it is never practicable to superheat to such a high temperature that the steam remains dry right to the end of the expansion, that is, right down to condenser pressure. Further, there is always a certain amount of water formed during the process of warming up and it is essential that this should not be permitted to accumulate, but should be blown through before the turbines are first put into motion. Many cases of failure of turbines can be traced to imperfect drainage, although, with care in design and in operation, there is little reason why these should occur

TEMPERATURE VARIATIONS AND HEAT ABSORPTION OF INTERNAL-COMBUSTION ENGINE PISTONS.*

By MAX JAKOB.

In a previous article[1] the author has dealt with the importance in engineering of a knowledge of the thermal conductivity of metals. Since considerable uncertainty exists as to how this and other thermal properties of metals come into prominence and should be allowed for in engine design, two particular problems of the heating of internal-combustion engine pistons will be dealt with in the following article.

1. Piston Temperature Variation.

The first question to arise is, how does the temperature of the combustion side of the piston vary with the time? The temperature of the working gases varies during each working cycle (in four-stroke engines every two revolutions) between 1,000° and 2,000° and over. If we assume that the metal surface cannot respond to these variations it is quite possible for the piston surface exposed to the gases, but not immediately water cooled, to undergo a temperature variation of at least 100°. This should be synonymous with a continuous rapidly changing stress of the piston material through tension and compression, which carried on for a sufficient length of time could prove fatal, because the time mean temperature of the piston surface is quite high. In order to avoid large variations of piston temperature, superficial considerations similar to those by means of which H. Lorenz[2] has shown for a steam engine, show that the heat entering and leaving the cylinder cover is approximately four to seven times the steady heat flow. "Hence it follows that in practice one must devote much more attention to the periodic heat losses than to the external outflowing heat which depends essentially on the average temperature drop of the cylinder walls." In internal-combustion engines precisely the reverse is the case. In steam engines the heat transfer to the walls is so good that these actually follow the temperature variations of the steam. It is therefore probably sufficient to assume the temperature variations on the outer surfaces of the walls as approximately known quantities, as has been done by Lorenz and W. Hort[3]. With internal-combustion engines, on the contrary, on account of the poor heat transfer and powerful cooling, this is not permissible. In this case one must rather start from the temperature variations of the working gases, since these have been frequently measured of late[4]. The variation of the wall temperature can be calculated in the manner adopted by F. Krauss[5]. It is sufficient to discuss here the results of these calculations and to illustrate them.

If the temperature θ_a of the working medium changes with the time t according to

$$\theta_a = B \sin (ct) + E \quad \ldots\ldots(1)$$

where B, c and E are suitable constants, then the temperature at a depth x in the wall can be expressed :—

$$\theta = C + D_1 = A e^{-mx} \sin \left(2a^2 m^2 t - mx - \tan^{-1} \frac{\lambda m}{\alpha + \lambda m} \right) \ldots(2)$$

It is herein assumed that the heat enters perpendicularly to the walls, which is only approximately true of internal-combustion engine pistons and only holds good for small values of x. C is the time mean temperature of the inner piston surface, $D = \frac{\alpha}{\lambda}(C - E)$ and is the second coefficient of the constant part of the heat flow. This portion is greater the greater the value fo α for the heat transmission between the gas and the piston, and is smaller the smaller the value of the thermal conductivity λ of the piston material. The amplitude A of the temperature vibration at the inner surface of the piston is given by—

$$A = \frac{\alpha}{\lambda m} B \sin \tan^{-1} \frac{\lambda m}{\alpha + \lambda m} \quad \ldots\ldots(3)$$

It is thus proportional to the quantity $\frac{\alpha}{\lambda}$ and to B, the amplitude of the temperature variation of the gas. Further we have

$$m = \sqrt{\frac{c}{2a^2}} \quad \ldots\ldots(4)$$

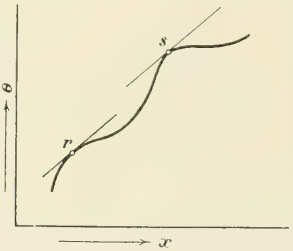


Fig. 1.—Curve showing Typical Local Distribution of Temperature in the x Direction.

Here a^2 is the so-called temperature conductivity which for variable heat flow is of the same importance as λ the thermal conductivity for constant heat flow. The quantity a^2 although well known in physics is not yet too well known in engineering. It is defined as

$$a^2 = \frac{\lambda}{\rho c} \quad \ldots\ldots(5)$$

where ρ is the density, c the specific heat, ρc thus characterising the specific heat per unit volume. The importance of a^2 is seen from the differential equation for the motion of heat in a given direction thus—

$$\frac{\partial \theta}{\partial t} = a^2 \frac{\partial^2 \theta}{\partial x^2} \quad \ldots\ldots(6)$$

The curve in Fig. 1 shows a typical local distribution of temperature in the x direction. The points r and s on the curve have the same temperature drop $\frac{\partial \theta}{\partial x}$. On the other hand the curvature at s is greater. Equation 6 thus implies that the temperature at s changes more quickly than at r. The purpose of the partial differential equation 6 is thus to obtain the time-temperature distribution from the space-temperature distribution. The equation also enables the rate of the temperature change at a given point, but with different materials, to be examined. If the temperature distribution curve of copper is examined it is seen that at any point the temperature changes roughly six times as quickly as with the same temperature distribution in bronze, because a^2 is about that much greater (*see* Table I).

From equation 6 and the more general equation

$$\frac{\partial \theta}{\partial t} = a^2 \left(\frac{\partial^2 \theta}{\partial x^2} + \frac{\partial^2 \theta}{\partial y^2} + \frac{\partial^2 \theta}{\partial z^2} \right) \quad \ldots\ldots(7)$$

the change of temperature with time can be graphically obtained for any point of the material once the space-temperature distribution is known. To accomplish this it is only necessary to differentiate the space temperature distribution

* Translated from the *Zeitschrift des Vereines deutscher Ingenieure* for Dec. 23, 1922.

TABLE I.—*Thermal Conductivity λ and Temperature Conductivity a^2 of various metals at 20° C. (approx.).*

Metal	λ = Cal. per sec. per sq. cm. per 1° C. per cm.	a^2 = sq. cm. per sec.
Silver	362	0·626
Copper	321	0·410
Aluminium	173	0·297
Platinum	51	0·067
Nickel	51	0·055
Iron:		
Mild Steel	45 to 48	0·052 to 0·056
Wrought Iron	40 to 54	0·047 to 0·062
Cast Iron	34 to 55	0·041 to 0·068
Steel	34 to 48	0·039 to 0·055
Constantan (60% Cu. 40% Ni.)	19	0·022
German Silver Alloy (54% Cu. 46% Ni.)	17	0·020

curves twice for each of the directions x, y and z, and by multiplication by a^2 the right-hand side of equations 6 and 7 is obtained. This simple application of equations 6 and 7 has not up till now very frequently been referred to. The integration of these expressions has been more often attempted, but, generally this step is both unnecessary and sometimes impossible.

The term e^{-mx} in equation (2) is the temperature vibration decremental factor in the wall. The amplitude of the temperature vibration is smaller the deeper the entry of the wave; and the greater c is, *i.e.*, the smaller the period of vibration and the smaller a^2 is. The remaining expression in equation 2, *i.e.*,

$$-mx - \tan^{-1}\frac{\lambda m}{a + \lambda m}$$

represents the phase difference between the temperature vibration in the walls and that in the combustion space.

For the temperature at the inner surface of the piston ($x = o$) we obtain from equations 2 and 3

$$\vartheta x = C + A \sin\left(2a^2m^2t - \tan^{-1}\frac{\lambda m}{a + \lambda m}\right)$$

$$= C + \frac{a}{\lambda m} B \sin\tan^{-1}\frac{\lambda m}{a + \lambda m} \sin\left(2a^2m^2t - \tan^{-1}\frac{\lambda m}{a + \lambda m}\right) \quad \ldots(8)$$

It is assumed in the above that the variation of ϑa can be expressed by a sine relation. However, from the researches of Petersen, Wolff and K. Neumann[7], it is found that the gas temperature changes according to a much more complicated law. This can be seen from an harmonic analysis of the temperature curves. For approximate calculations, however, the sine law will be sufficiently correct. Moreover, the heat transmission coefficient a is assumed to be constant in the above whilst the coefficients obtained by Neumann changed during each double stroke, and those obtained by W. Nusselt[8] changed decidedly with the time. Further, Wolff has shown that the temperatures at various points of the combustion space of a Diesel engine differ greatly. Finally, the piston surface has a marked radial temperature drop; this fact has been pointed out by W. Riehm[9] from experiments on a Diesel engine, and by G. Becker[10] from experiments on a tractor motor. All these space and time inconsistencies can be neglected in the following approximate calculations. These only deal with the magnitude of the amplitude A of the temperature vibration about the mean temperature C of the inner surface of the piston.

The piston is formed of cast-iron of $\lambda = 50$, $a^2 = 0{\cdot}055$. Let the instantaneous maximum temperature $(\vartheta_a)_h$ of the gases be 1,500°, 2,000°, and 2,500° C., the corresponding lowest temperatures $(\vartheta_a)_l$ being 100°, 150° and 200°; therefore $B = \frac{(\vartheta_a)_h - (\vartheta_a)_l}{2}$ = 700°, 925° and 1,150°. For these three cases we can assume from Neumann's work, the following values for α: 40, 70 and 100, these coefficients being for a reddish black surface. The shaft of the four-stroke machine making $n = 125$, 250, 500, 1,000 and 2,000 revs. per minute, the hourly periodicities of the assumed sine vibration is therefore $\frac{c}{2\pi} = 3{,}750$, 7,500, 15,000 30,000 and 60,000, and hence $m = 463$, 654, 926, 1,309 and 1,852. For $\sin\tan^{-1}\frac{\lambda m}{a + \lambda m}$ we can assume, since a is always small compared with λm, that $\frac{\lambda m}{a + \lambda m}$ equals 1, and therefore we have[16] $\sin\tan^{-1} 1 = 0{\cdot}7071$. Therefore the required amplitude A equals $0{\cdot}7071 \frac{a}{\lambda m} B$. From this we obtain the values given in Table II.

TABLE II.—*Amplitude A of the Temperature Vibration at the inner surface of an Iron Piston.*

n =		125	250	500	1,000	2,000
$(\vartheta a)_h$	$(\vartheta a)_l$					
1,500	100	0·8	0·6	0·4	0·3	0·2
2,000	150	2·0	1·4	1·0	0·7	0·5
2,500	200	3·5	2·5	1·8	1·3	0·8

For aluminium pistons of $\lambda = 180$, $a^2 = 0{\cdot}30$ the values of A in Table II should be multiplied by

$$\frac{\lambda}{a\,a}\text{ iron} \div \frac{a\,a}{\lambda}\text{ aluminium.}$$

Assuming blackened surfaces in both cases and α the same for each, we have

$$\frac{50}{\sqrt{0{\cdot}055}} \div \frac{\sqrt{0{\cdot}30}}{180} = 0{\cdot}649.$$

The amplitude of temperature vibration in aluminium pistons is thus roughly $\frac{2}{3}$ of that for iron pistons. Although aluminium conducts heat 3½ times better than iron, very little difference thus exists between their respective temperature vibrations. In each case, however, the temperature variation of the inner surface of the piston is quite small. From Table II it is seen that the variation in slow running engines with a maximum temperature of 2,500° only amounts to about ± 3·5° with quick running engines, lower temperatures and aluminium pistons a correspondingly lower variation is obtained. Even were the value of a doubled (for example, Schlachter[11] has used a value of $a = 192$), the temperature variation is still appreciably less than ± 10°.

II. Heat Absorption of Pistons.

Up till now we have taken the same value of a for both iron and aluminium with blackened surfaces. The head absorption, however, is in general dependent upon the nature of the absorbing material as has been recently shown by Becker and A. Rieder.[12] This influence is unimportant for small temperature variations of the piston, but is very important in the loss due to constant heat flow. The differences in the heat absorption of the various metals which were examined by Becker originate essentially only from the heat transferred by radiation. A blackened aluminium piston has approximately the same value of a as has an iron piston. On the other hand, a polished copper or aluminium piston absorbs materially

less heat then a cast-iron piston. If two parallel plates of area F and temperatures T_1 and T_2 are so arranged that loss of heat at the boundaries of the plates is avoided, then the first plate receives through radiation from the second, an amount of heat equal to

$$W = \frac{F}{\frac{1}{c_1} + \frac{1}{c_2} - \frac{1}{c}} (T_1^4 - T_2^4) \quad \ldots\ldots\ldots\ldots (9)$$

Where C_1 and C_2 are constants, depending upon the material of the plates, and $C = 4{\cdot}90$, 10^{-8} κ cal. m^{-2} h^{-1} degrees^{-4} is the constant in the Stefan-Boltzmann law of radiation. If the first plate is imagined to be taken away, and in its stead an infinitely thin gas film substituted of temperature T_1 which has absorption power but no noticeable power of reflection, this film thus radiates—assuming the validity of the Kirchhoff law of radiation as a blackened body of the same temperature.[13] Therefore $C_1 = C$ and equation 9 thus becomes

$$W = C_2 F (T_1^4 - T_2^4) \quad \ldots\ldots\ldots\ldots (10)$$

This expression represents the heat given up by radiation of an infinitely thin gas film of temperature T_1 to a plate of temperature T_2. The constants in this equation have been measured by Wamsler[14] for various materials, and are given in Table III.

TABLE III.—*Constant C_2 of Heat Absorption through Radiation (Wamsler).*[15]

Material.	Surface.	Plate Temperature (°C.	$10^8 \cdot C_2$ κ cal. m^{-2} h^{-1} degrees^{-4}.
Cast-iron	Rough, oxydised	40 to 250	4·41
Lamp black	Smooth	0 to 50	4·36
Malleable iron	Dull, oxydised	20 to 360	4·32
Lime mortar	Rough white	10 to 90	4·22
Malleable iron	Highly polished	40 to 250	1·32
Brass	Dull	50 to 350	1·03
Zinc	Dull	50 to 290	0·97
Copper	Highly polished	50 to 280	0·79

Becker's observation that rusted piston heads absorb on the average 2·6 times as much heat as do polished, agrees fairly well with Table III. It is therefore important, what, of course, was previously clear from physical considerations, to have polished surfaces in internal-combustion engine cylinders. Since, from Becker's observations, light metal pistons are found to be susceptible to rust formation, which in service rapidly exfoliates, the heat loss of a motor with light metal pistons, even with the same temperature of inner surface, is smaller than in cast-iron pistons. Unfortunately, the researches of Becker do not admit of further conclusions being drawn as to the heat absorption, because the heat was radiated not from a gas but from a body of unknown temperature, which also radiated to the mercury thermometer. One method of determining the radiation of a gas of finite thickness has been shown by Nusselt in the previously-mentioned work. He carried out experiments on a bomb calorimeter whose inner surface was first blackened and next polished and thus determined the absorptive power of the inner surface. Further research in this direction is very desirable.

From the foregoing considerations which show the temperature variations in internal-combustion engine pistons to be so small, it appears that more accurate research does not seem to be urgently desirable. It would be much more desirable to measure the gas temperature variations. With regard to pistons, research is particularly necessary on the local or space-distribution and the time-average value of the temperature, together with measurements of heat transmission in hollow spheres and cylinders, dealing particularly with the radiation portion of the heat absorbed by different materials. Such research would greatly extend our knowledge of the heat distribution in the cylinders of internal-combustion engines.

FOOTNOTES AND REFERENCES.

1. See *Z. d. V. d. I.*, 1922, page 688.
2. *Lehrbuch der techn. Physik.* Volume II., 1904, page 458.
3. *Die Differentialgleichungen des Ingenieurs*, 1914, page 328.
4. A. Petersen, *Forschungsarbeiten*, 1913. No. 143, page 37, also E. B. Wolff, *Der Olmotor*, 3, 1914-15, page 451; 4, 1915-16, pages 9 and 52.
5. *Die Grundgesetze der Warmeleitung und ihre Anwendung auf platten förmige, Körper. Z. d. V. d. I.*, 1919, page 64.
6. From Landolt-Bornstein, *Physik-Chem. Tabellen*, 4 Edition. The values for iron from W. Beglinger, *Verhandl d. Ver zur Beförderung des Gewerbfleisses*, 75, 1896, page 32.
7. *Z. d. V. d. I.*, 1921, page 801.
8. *Z. d. V. d. I.*, 1914, pages 361 and 520.
9. *Z. d. V. d. I.*, 1921, page 923.
10. *Vervollkommnung der Kraftfahrzeugmotoren durch Leichtmetallkolben*, 1922. *Z. d. V. d. I.*, page 343.
11. *Motor and Auto.* 1922, pages 97, 113, and 129.
12. *Z. d. V. d. I.*, 1922, page 343.
13. A simple proof has been given by Kurlbaum. See F. Henning, *Temperaturmessung, Braunschweig*, 1915, page 198.
14. *Forschungsarbeiten*, Heft 98 and 99, 1911. *Z. d. V. d. I.*, pages 599 and 628.
15. Revised and improved by the author.
16. *Translation Note.*—The expression, Sin tan^{-1} 1, was expressed in the original as Sin arc tang 1, and was said to equal 0·8415. This is evidently an error on the author's part. Table II given here has, therefore, been corrected.

MOTOR SHIPBUILDING ON THE CLYDE.

Out of 108 vessels which were under construction in Clyde shipyards—either fitting out in the harbour or on the stocks at the end of March last, 11 are motorships, which, when constructed, will have an aggregate tonnage of 76,825. Since the end of March, however, the number has been substantially increased by the placing of orders for five motorships with two firms—Harland & Wolff and Lithgows Limited. On the East Coast of Scotland there is one motorship of 7,700 tons being built at Dundee, and also a small motor vessel of about 140 tons at Aberdeen. Of the motorships which are now nearing completion in Glasgow Harbour the most notable is the *Dalgoma*, which A. Stephen & Sons are building for the British India Company, and this vessel was due to run trials last month. Harland & Wolff have completed, at Govan, the *Ediba*, and launched another large motor vessel. The *Ediba*, which is a vessel of 6,750 tons, has been built for Elder Dempster & Co., Liverpool. The other motorship has been named the *Lugma*, and she is owned by the Pacific Steam Navigation Company.

MECHANICAL GEARING.*

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The progress of the marine steam turbine has, as a result of the rapid developments of the last ten years, been accompanied with that of the means of transmission of power to the slow moving propeller. Of the various means of transmission, mechanical gearing on account both of its simplicity and of its efficiency has been most widely adopted. Introduced primarily for the purpose of extending the field of utility of the turbine to the propulsion of low-speed mercantile vessels, for which the revolutions of the propeller were too low to make direct-coupled turbines an economical proposition, the advantages of freeing the turbine from the limitations to its revolutions imposed by the propeller was soon recognised to be of value even in those fields of work which were then open to the direct-coupled turbine. It has consequently been almost completely displaced in naval and high-speed mercantile work throughout the world by the geared turbine. Since their first experimental adoption in the *Vespasian*, a description of which was given in papers before the Institution in 1911 and 1912, the output of mechanically geared marine turbines has grown rapidly, and the total output has equalled and even surpassed that of direct-coupled turbines. Of the total output in British-built vessels, about 10,000,000 h.p. has been transmitted with a single reduction in speed, and about 1,000,000 h.p. with a double reduction. Single reduction is chiefly used in naval work. It was recognised from the first that there were circumstances in a marine drive which made necessary a considerable margin of strength over and above what was calculated for the normal power transmitted, to cover possible inaccuracies in the gear cutting, lack of exact alignment, and shocks due to rough seas or even the striking of the propeller blade against a submerged obstacle, and to meet these contingencies such a margin was provided. The first gears were regarded as experimental, and increase to larger powers was made in cautious stages. The demand for quick production during the war led to an abandonment of some of this caution, but the record of geared turbines in the Navy during the war has shown that this confidence in mechanical gearing was not misplaced. In some instances the power successfully transmitted through a single pinion has been as high as 20,000 h.p., with tooth pressures as high as 1,100 lb. per in. of surface. The gears for naval work have been almost exclusively single reduction, usually consisting of two pinions engaging with a common

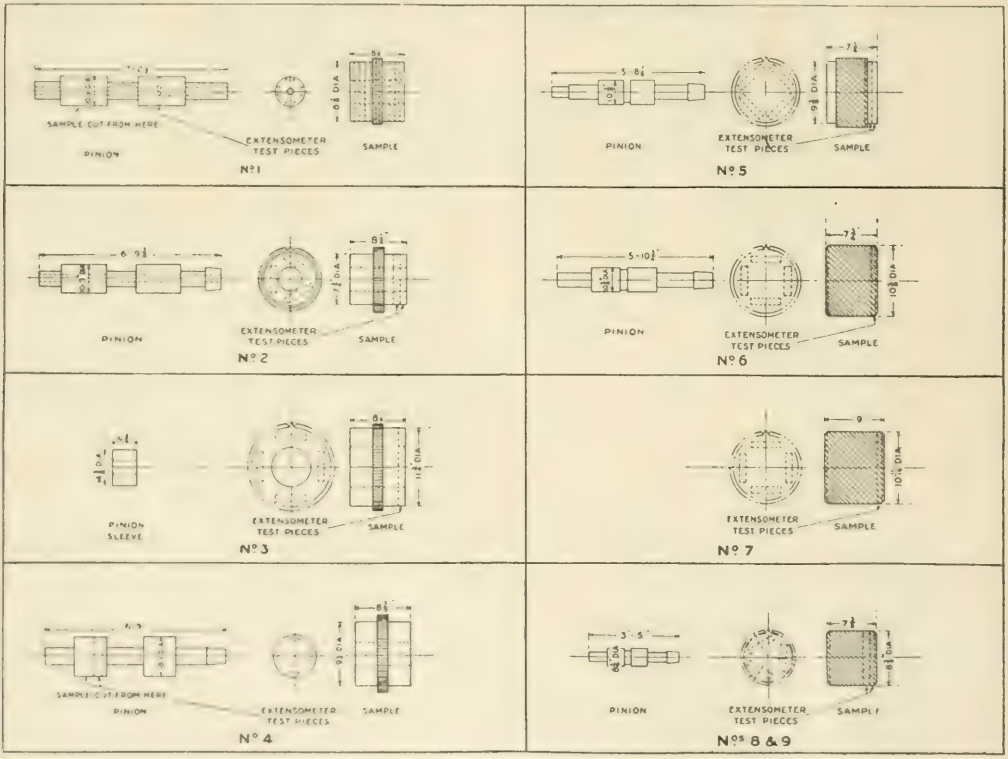

Fig. 1.—Dimensions of Reduction-Gear Pinions and Sample Pieces.

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wheel on the propeller shaft. Where, however, geared cruising turbines have been fitted, the power of the cruising turbine has frequently been transmitted by double reduction, the cruising turbine being connected through a single wheel and pinion to the shaft of one of the main turbines, itself geared to the propeller shaft. The largest ratio of reduction adopted with a single step from turbine to propeller shaft was 26 to 1 in the case of the *Cairnross*, a mercantile vessel built in 1912 which was sunk during the war. It was soon recognised that in mercantile work, to obtain full advantage of the economy of high-speed turbines, a still larger ratio of reduction was required, and this led to the adoption of double-reduction gears. The development of mercantile work was, however, interrupted by the war, and it was not until 1918 that the first marine double-reduction gear was fitted in this country in the ss. *Somerset*. This gear had teeth of $\frac{1}{12}$-in. pitch in the primaries with a load of 560 lb. per in., and teeth of 1-in. pitch in the secondaries with a load of 982 lb. per in., and this vessel has run a total distance of about 170,000 miles. Reference should, however, be made to a turbine-driven rolling mill fitted with double-reduction gear which was installed at Calderbank in 1910, and has run satisfactorily up to the present time. The load on the secondary teeth of this gear is 1,150 lb. per in. Since the time when the *Somerset* was placed on service there have been about 207 British-built vessels fitted with double reduction, and during the same period 26 with single reduction. Unfortunately, the excellent record established by the earlier gears has not been maintained. Commencing in December, 1921, there followed throughout last year repeated instances of failure of newly installed gearing owing to the fracture of pinion teeth. There has been a general tendency to associate these failures specially with double reduction, but in reality the percentage of failures is no more in double than in single reduction in mercantile work during the same period. Out of 207 vessels with double reduction there have been 16 such cases of failure, and in single reduction four cases of out 26. So that whilst it may be recognised that the effect of inaccuracies in the teeth is likely to be greater in double reduction on account of the greater energy of the high-speed turbines, this cannot, in face of the facts, be accepted as the explanation of these failures. The design of the single-reduction gears, too, must be acquitted when we take into account the mass of successful naval work with gears of similar type. When the report came to the second of these failures by fracture, the authors and their colleagues immediately took up with the steel makers the question of the quality of the steel. Investigation of the physical characteristics and both macroscopical and microscopical examination of nickel steel pinions which had failed in this way showed considerable variation in grain size and in physical properties, but did not supply a sufficient explanation for these failures. In almost all cases the fractures were what are termed creeping fractures, consisting of a fine-grained area and a smaller coarse-grained area where the teeth finally broke away. Recent experiments by Dr. Hatfield have shown that such fractures can be produced by repeated stresses in one direction and well above the elastic limit. In the very early experiments which led to the introduction of mechanical gearing, after the original gearing of the *Vespasian* had been well tried out, the original pinion was replaced by one of high tensile steel, with the result that several of the pinion teeth fractured on the first voyage. From this experience it was concluded that as a necessary requirement the steel must possess a high degree of ductility. As confidence grew, however, this rule has not been rigorously observed. The tendency has been to use harder steel, nickel steel, or chrome nickel steel, hardened by oil quenching, and it is with such steels that practically all the fractures have occurred. Investigations have been made by the authors and their colleagues into the possibilities of producing cracks in oil-quenched nickel steel by heat treatment, such as might be employed in the ordinary process of manufacture. The nine samples experimented upon were all portions of turbine reduction gear pinions of cylindrical shape from 8 in. to 12 in. diameter and from 8 in. to 10 in. long. The dimensions of the samples, and of the finished pinions from which the samples were cut, are shown in Fig. 1. It will be observed from these that some of the pinions have central holes. Except in the case of sample No. 4 these were bored and all rough machining done before oil quenching by the makers. The makers' heat treatment of the forging from which each sample was taken, where known, is stated in Table I.

Table I.

Records of Heat Treatment by Makers.

Sample No.	Quenched at ° C.	Tempered to ° C.
1	770	620
2	850	600
3	870	650
8	870	650
9	870	650

Six of these pinions had actually been for some time in service. Of the three others, one was rejected through an error in machining and two were replaced through a change in design. The material in all cases was nickel steel to the usual specification, viz., carbon 0·25 to 0·32, nickel about 3½ per cent. The heat treatment in these experiments consisted in heating the sample to a high temperature in a gas-fired furnace, and either quenching it in oil or allowing it to cool out in the atmosphere on a steel plate on the shop floor; the procedure generally followed was to subject the samples to a series of such treatments alternately, quenching from 850° C., reheating to 600° C. for tempering, and cooling in air. The initial temperature of the furnace on the insertion of the sample was varied, and in some instances the sample was enclosed in a muffle. Temperatures were observed by two pyrometers, one of the ordinary thermocouple type in which the couple was inserted in the furnace and therefore gave the temperature of the furnace itself. The other a radiation pyrometer which was focused on the sample or enclosing muffle and therefore indicated the temperature of the latter. The accuracy of these instruments was verified during and after concluding the series of experiments.

After each treatment the samples were examined for cracks, and to facilitate this the plain surfaces were in the majority of cases ground or polished. The following table gives a general summary of the experiments :—

Table II.

Sample No.	Date of Manufacture.	Percentage of Nickel Content Specified.	No. of Times Quenched.	No. of Times Tempered.	Cracks Developed.
1	1917	3¼ to 3¾	2	1	Yes
2	1917	3½ to 5	5	5	No
3	1911	3½	4	4	No
4	1912	3½	17	16	No
5	1920	3½ to 5	12	13	No
6	1920 or 1921	3	7	7	No
7	1921		12	12	No
8	1919	3½ to 5	8	8	Yes
9	1919	3½ to 5	8	8	Yes

Tables IV to XI of the Appendix show the detailed records of the heat treatment of the various samples. Sample No. 1 was from a pinion which was spoiled through an error in machining, but which would otherwise have gone into service since it had passed the prescribed mechanical tests. This sample was subject to two quenchings and one tempering as given in Table IV. After the second quenching it was found to have developed a large number of cracks on the flat ends. The cracks can be seen in the photographs, Fig. 2 (Plate II). The

sample was then annealed and cut up for examination, as explained by the dotted lines in Fig. 1, and photographs Figs. 2 and 3, show the cracks as they appeared in the various sections. Samples No. 8 and No. 9 were from a high-pressure pinion of a double-reduction set of gears, one sample being taken from each helix. This pinion had been on service about six months but had not failed or been damaged in any way beyond a certain amount of pitting. The low-pressure pinion, however, which was from the same maker and presumably from the same cast, had failed by tooth fracture and was re-sleeved. It was therefore considered advisable to replace the high-pressure pinion also, and this left the original

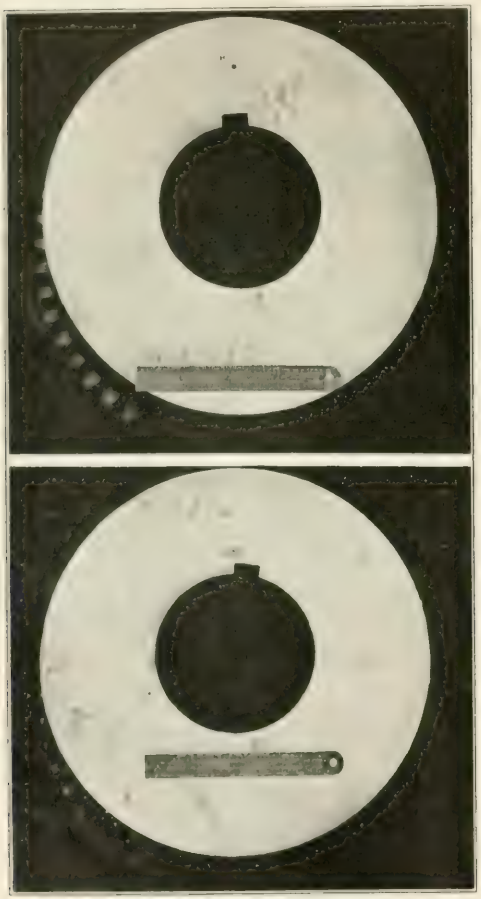


Fig. 2.—Pinion subjected to Heat Treatment

pinion available for investagtion. The samples Nos. 8 and 9 were subjected together to six quenchings and eight temperings. Three or four cracks appeared on one end of sample No. 8 after the third quenching. One of these was radial and the others principally radial. For the next heat treatment these two samples were heated in the furnace at a temperature of 600° C. and then allowed to cool out in the atmosphere, when two additional cracks were found to have developed in No. 8 and two radial cracks in one end of sample No. 9. The cracks in sample No. 8 are shown in the photograph, Fig. 5. After the seventh quenching another small crack appeared at the other end of sample No. 8 and two more cracks in sample No. 9. Thus cracks have been produced in three samples out of nine submitted to experiment by such heat treatment as nickel steel forgings are liable to be subjected to in the course of manufacture. Some of these nine samples were from pinions which had failed by tooth fracture, but it should be observed that the three in which cracks were produced had not actually broken, one of them never having been in service. It should be observed, also, that all these pinions had passed their specified mechanical tests, but obviously test-pieces afford no indication of the state of internal stress. Fig. 6 is a series of micro-photographs from sample No. 1: A before the experimental heat treatment, B from portion A9 (Fig. 4) away from the cracks, and C from the same portion showing cracks. With regard to the last two, it should be stated that the sample from which they are taken had been annealed before cutting up. Micro-photographs from sample No. 4 are shown in Fig. 7: D as it was after having undergone only the makers' heat treatment, E and E1 after the series of quenchings and temperings recorded for that sample. Extensometer tests have been made with transverse test-pieces for each of these samples, in the case of Nos. 1 and 4 from another part of the body of the same forging, in the other cases from the samples themselves after the experimental heat treatment.

The results are given in Table III and show considerable variation in proportionality limit, elongation, and reduction of area. The heat treatment of nickel steel forgings is liable to produce in them strains of extreme severity. The exact nature of the strains it is not easy to determine. Were it only a question of simple expansion or contraction with varying temperature, the sudden cooling of the outer surface while the core is still hot will put that surface for the moment into a state of tension which the subsequent contraction of the core would reverse, and ultimately there would be compression strain on the surface and tensile strain in the interior. The action, however, is complicated by the changes of volume at recalescence points, first beneath the cooled surface and then gradually farther in, and varying strains and stresses are set up in all parts of the pinion. Heavy stresses may thus come upon some portion of the steel when it is at such a temperature as to be in a brittle condition and result in a crack or in permanent injury to the steel.

The conclusion of the authors is that oil quenching of large masses of nickel steel introduces risks of cracks and excessive strains, and should be discontinued for turbine gears. The wheel rims are not made of oil-quenched steel, and the freedom from fracture of wheel rim teeth, and of those of pinions which have been normalised only, confirms this conclusion.

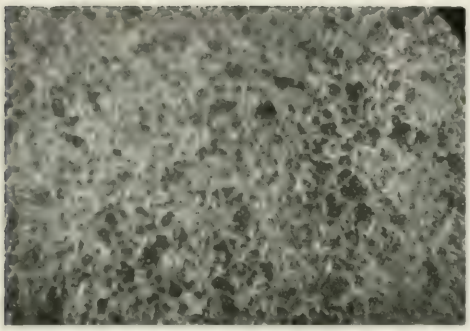
With steel in a homogeneous unstressed condition the factor of safety is a large one. Reference was made to this in the reply to the discussion on Walker & Cook's paper in 1921. In an experiment, static loading 40 to 100 times in excess of the normal tooth pressure failed to break the teeth. The authors have in hand at the present time a further experiment to determine the maximum safe load per inch under steady running conditions; but in the early experiments made prior to the installation of geared turbines in the *Vespasian*, to determine the load at which destructive wear begins to take place, the limit was found to be about 1,200 lb. per in. with a 5-in pinion, or 240 times the diameter of the pinion in inches. This limit varies directly as the diameter of the pinion. The foregoing consideration of these matters does not imply a neglect of others which are recognised to be of great importance for the successful running of mechanical gears—chief of these is the question of accurate machining, in particular the cutting of the teeth. The principal effect of an error in the cutting of the teeth is as follows: With two pinions driving one wheel, unless the errors on the two sides synchronise, it is possible for

Fig. 3.

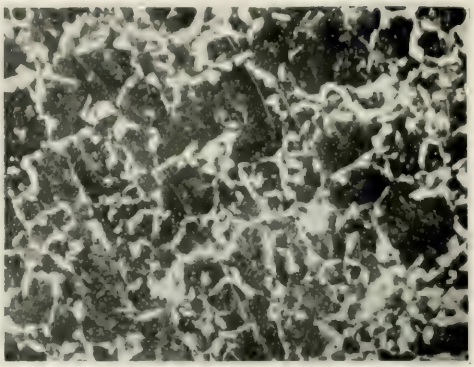
Fig. 4.

Fig. 5.

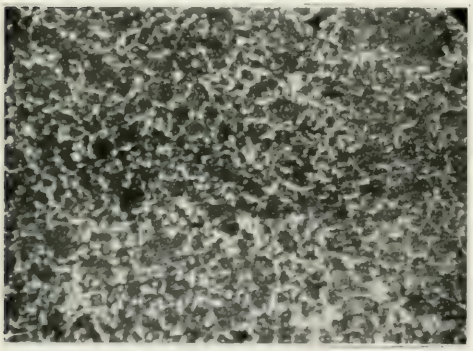
PHOTOGRAPHS SHOWING CRACKS WHICH DEVELOPED AS A RESULT OF HEAT TREATMENT.



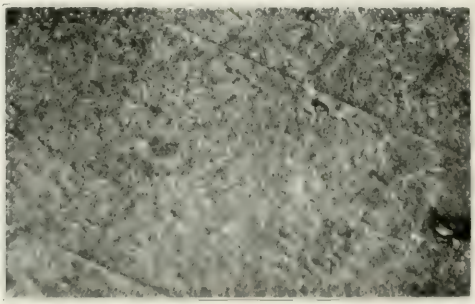
A. Sample No. 1. Maker's Heat Treatment.



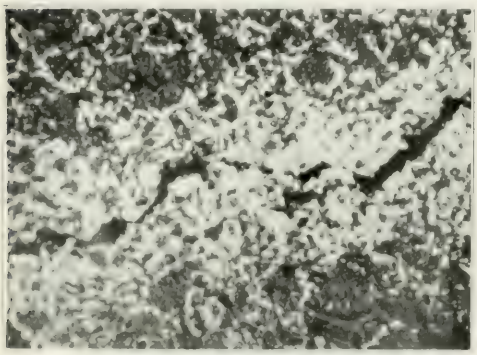
D.—Sample No. 4. Maker's Heat Treatment.



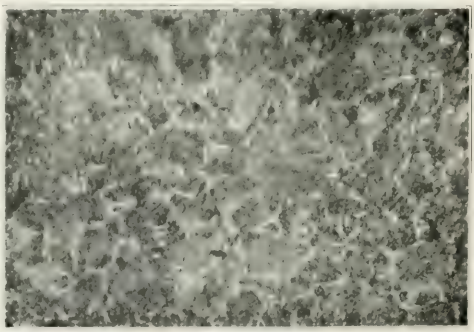
B.—Sample No. 1. After Experimental Heat Treatment.
(Section A 9 of Fig. 4.)



E.—Sample No. 4. After Experimental Heat Treatment.



C. Sample No. 1, with Crack Developed.
(Section A 9 of Fig. 4.)



E 1.—Sample No. 4.—After Experimental Heat Treatment.

Figs. 6 and 7. MICRO-PHOTOGRAPHS SHOWING RESULTS OF VARIOUS HEAT TREATMENTS.

one pinion to be tending to push the wheel ahead, and for the other to hold back, and there is thus a state of war between the pinions with the inertia of the turbines behind them striving to keep them running at constant speed. To overcome this it has been proposed in some cases, where errors are known to exist in the cutting machine, to mark the pinions and primary wheels in reference to their position on the machine when cutting, and subsequently in assembling to "register" the

TABEL III.—*Extensometer Tests.*

Sample No.	Elastic Limit.	Yield Point.	Maximum Stress.	Elongation, Per cent. (2 in.).	Reduction of Area, Per cent.	
1	25·0	36·6	46·2	21	36·4	
1A	26	30	46	24	42·7	F.*
1B	26	30·8	45·7	25	41·2	F.
1C	19·7	29·3	34	3	2·8	g.
2	19·6	41·6	47·1	21·5	37·5	
2A	13·5	29·7	47·1	20	30·6	F.
2B	13·5	32·5	48·75	15	23·3	F.
2C	14·06	32·5	48·5	15	33·4	F.
3	26·9	31·7	47·1	18·5	39	F.
3A	24·1	29·7	45·2	19·5	41·4	F.sl
3B	28·4	32·4	47·6	20	38	F.sl.
3C	28	32·4	47·8	19	34·6	F.
4	18·6	37·9	44·6	20	33	
4A	19·3	24·9	45·5	12·5	11·6	g.
4B	15·65	24·5	45·2	12·5	11·6	g.
4C	18·0	24·5	45	10·5	13·7	g.
5	26·8	37·6	45·2	19	27·6	
5A	22·5	28·9	45·5	23	38·2	F.sl.
5B	26·1	30·1	46·3	21	38·1	F.sl.
6	31·6	42·7	50·6	15	26·7	
6A	30·5	40·5	51·1	15	30·5	F.sl.
6B	28·5	39·7	51·8	12·5	20·4	F.sl.
6C	32·2	43·5	53·7	12·5	17·5	F. g.
7	32·0	43·4	57·6	18	48	
7A	—	43·4	52·5	19·5	53·8	F.
7B	34·8	44	53·8	18·5	46·4	F.
7C	30·1	39·4	50·7	21	51	F.
9	17·4	31·6	40·7	24·3	37	
9A	18·4	25·6	41·32	22·5	36	F.
9B	19·3	27·7	42·3	18·5	35	F.

* Fractures :—F. = fibrous ; g = granular ; sl. = silky.

pinions and primary wheels on one side with those on the other, so that whatever errors of angular position may come to them from the gear-cutting machines may synchronise. This has actually been done in a few instances with marked success, as judged by quietness of running. It does not, however, counteract errors in the secondary wheel A better proposal, however, is to insert flexibility combined with damping between the main wheel and the turbines, or at any rate between the turbines. A method of doing this was described during the discussion on Dr. Smith's paper last spring. By the courtesy of Alfred Holt & Co. it is being fitted to one set of gears of a twin-screw vessel which will shortly go into service, and actual experience of such a provision will soon be available. With devices of this kind a larger error in the gear teeth can be tolerated. Without them, a higher standard of accuracy is required to obtain quiet running gears, and it is appreciated that no effort should be spared in the improvement of gear-cutting machinery. At the same time, the gear-cutting machines as they are at present must be credited with a very high percentage of successful gears, and in the authors' opinion most of the instances of gears being unsatisfactory, through noise, for instance, should be attributed to want of care either in overhauling the machine or in setting the work rather than to inherent errors in the master wheels of the machines. Perfect accuracy is an ideal. The question may be approached from two standpoints : How far is it necessary to go ? In other words, What is the maximum error that can be allowed ? Or, how far is it possible to go in the direction of accuracy ? What is the minimum of error that can be attained ? To this latter question the Parsons Marine Steam Turbine Company are giving very close attention, and have in hand at present improvements which will further increase the accuracy of their machines, and by these means it is hoped to reduce the maximum error of angular position for large wheels to about $\frac{1}{20000}$ in. Regarding this as the accumulated error due to variation in pitch, it will represent an individual pitch variation (which is frequently adopted as the measure of accuracy) of the order of $\frac{1}{1000000}$ in. Two lines of advance are being followed, one involving a minute rectification of master wheels, while the other is the development of an improved creeping mechanism.

APPENDIX.

TABLE IV.

Record of Experimental Heat Treatment of Sample No. 1.

Date.	Temperature of Furnace *before* Insertion.	Temperature of Furnace *after* Insertion.	Maximum Temperature.	Cooled out in	Cracks Developed.	Muffle.
	° C.	° C.	° C.			
20/7/22	20	20	830	Oil	—	No
1/8/22	580	400	830	Oil	Cracked on ends	No.
25/9/22	20	20	850	Furnace (Cut up and examination)		for ex-

TABLE V.

Record of Experimental Heat Treatment of Sample No. 2.

Date.	Temperature of Furnace *before* Insertion.	Temperature of Furnace *after* Insertion.	Maximum Temperature.	Cooled out in	Cracks Developed.	Muffle.
	° C.	° C.	° C.			
20/7/22	18	18	830	Oil	None	No
2/8/22	500	350	830	Air	None	No
4/8/22	18	18	845	Oil	None	Yes
10/8/22	20	20	400	Air	None	Yes
12/8/22	20	20	850	Oil	None	Yes
22/9/22	600	500	600	Air	None	No
26/9/22	600	500	850	Oil	None	Yes
28/9/22	600	500	600	Air	None	No
29/9/22	600	500	850	Oil	None	No
3/10/22	600	500	600	Air	None	No

TABLE VI.

Record of Experimental Heat Treatment of Sample No. 3.

Date.	Temperature of Furnace *before* Insertion.	Temperature of Furnace *after* Insertion.	Maximum Temperature.	Cooled out in	Cracks Developed.	Muffle.
	° C.	° C.	° C.			
20/7/22	18	18	830	Oil	None	No
31/1/23	600	480	600	Air	None	No
2/2/23	600	480	850	Oil	None	No
2/2/23	600	480	600	Air	None	No
3/2/23	600	480	850	Oil	None	No
5/2/23	600	480	600	Air	None	No
6/2/23	600	480	850	Oil	None	No
6/2/23	—	—	600	Air	None	No

TABLE VII.

Record of Experimental Heat Treatment of Sample No. 4.

Date.	Temperature of Furnace *before* Insertion.	Temperature of Furnace *after* Insertion.	Maximum Temperature	Cooled out in	Cracks Developed	Muffle.
	°C.	°C.	°C.			
20/7/22	20	20	830	Oil	None	No
3/8/22	20	20	840	Oil	None	Yes
5/8/22	20	20	600	Air	None	Yes
8/8/22	20	20	850	Oil	None	Yes
9/8/22	20	20	600	Air	None	Yes
10/8/22	20	20	850	Oil	None	Yes
11/8/22	260	120	600	Air	None	Yes
14/8/22	150	—	850	Oil	None	Yes
16/8/22	20	20	600	Air	None	Yes
17/8/22	20	20	850	Oil	None	Yes
18/8/22	20	20	600	Air	None	Yes
19/8/22	20	20	850	Oil	None	Yes
21/8/22	20	20	600	Air	None	Yes
22/8/22	20	20	850	Oil	None	Yes
23/8/22	20	20	600	Air	None	Yes
24/8/22	20	20	850	Oil	None	Yes
25/8/22	20	20	600	Air	None	Yes
28/8/22	20	20	850	Oil	None	Yes
29/8/22	20	20	600	Air	None	Yes
30/8/22	20	20	850	Oil	None	Yes
31/8/22	20	20	600	Air	None	Yes
1/9/22	20	20	850	Oil	None	Yes
4/9/22	20	20	600	Air	None	Yes
6/9/22	20	20	850	Oil	None	Yes
8/9/22	20	20	600	Air	None	Yes
11/9/22	20	20	850	Oil	None	Yes
14/9/22	20	20	600	Air	None	Yes
15/9/22	20	20	850	Oil	None	Yes
19/9/22	20	20	600	Air	None	Yes
21/9/22	600	500	850	Oil	None	Yes
26/9/22	600	500	600	Air	None	No
27/9/22	600	500	850	Oil	None	No
29/9/22	600	500	600	Air	None	No

TABLE VIII.

Record of Experimental Heat Treatment of Sample No. 5.

Date.	Temperature of Furnace *before* Insertion.	Temperature of Furnace *after* Insertion.	Maximum Temperature.	Cooled out in	Cracks Developed.	Muffle.
	°C.	°C.	°C.			
11/8/22	20	20	850	Oil	None	Yes
14/8/22	20	20	600	Air	None	Yes
15/8/22	100	—	850	Oil	None	Yes
16/8/22	100	—	600	Air	None	Yes
17/8/22	200	—	850	Oil	None	Yes
18/8/22	200	—	600	Air	None	Yes
21/8/22	300	—	850	Oil	None	Yes
22/8/22	300	200	600	Air	None	Yes
23/8/22	400	310	850	Oil	None	Yes
24/8/22	400	310	600	Air	None	Yes
25/8/22	400	400	700	Air	None	Yes
26/8/22	400	380	850	Oil	None	Yes
28/8/22	400	400	600	Air	None	Yes
29/8/22	600	—	850	Oil	None	Yes
30/8/22	600	490	600	Air	None	Yes
31/8/22	700	—	850	Oil	None	Yes
1/9/22	700	580	600	Air	None	Yes
4/9/22	600	490	850	Oil	None	Yes
6/9/22	600	500	600	Air	None	No
8/9/22	600	500	850	Oil	None	Yes
11/9/22	600	500	600	Air	None	No.
14/9/22	600	500	850	Oil	None	No
15/9/22	600	500	600	Air	None	No
19/9/22	600	500	850	Oil	None	No
20/9/22	600	500	600	Air	None	No

TABLE IX.

Record of Experimental Heat Treatment of Sample No. 6.

Date.	Temperature of Furnace *before* Insertion.	Temperature of Furnace *after* Insertion.	Maximum Temperature.	Cooled out in	Cracks Developed	Muffle.
	°C.	°C.	°C.			
19/9/22	20	20	850	Oil	None	Yes
21/9/22	20	20	600	Air	None	Yes
22/9/22	600	500	850	Oil	None	Yes
27/9/22	600	500	600	Air	None	No
28/9/22	600	500	850	Oil	None	No
3/10/22	600	500	600	Air	None	No
4/10/22	600	500	850	Oil	None	No
6/10/22	600	500	600	Air	None	No
5/12/22	600	500	850	Oil	None	No
6/12/22	20	20	600	Air	None	No
8/12/22	600	500	850	Oil	None	No
12/12/22	20	20	600	Air	None	No
13/12/22	600	480	850	Oil	None	No
14/12/22	20	20	600	Air	None	No

TABLE X.

Record of Experimental Heat Treatment of Sample No. 7.

	°C.	°C.	°C.			
11/11/22	600	490	850	Oil	None	No
14/11/22	20	20	600	Air	None	No
17/11/22	600	500	850	Oil	None	No
22/11/22	20	—	600	Air	None	No
24/11/22	600	500	850	Oil	None	No
29/11/22	20	20	600	Air	None	No
6/12/22	600	500	850	Oil	None	No
8/12/22	20	—	600	Air	None	No
11/12/22	600	500	850	Oil	None	No
12/12/22	20	20	600	Air	None	No
14/12/22	600	490	850	Oil	None	No
15/12/22	20	20	600	Air	None	No
21/12/22	600	500	850	Oil	None	No
3/1/23	600	480	600	Air	None	No
9/1/23	600	470	850	Oil	None	No
10/1/23	600	480	600	Air	None	No
11/1/23	600	480	850	Oil	—	No
12/1/23	800	700	600	Air	None	No
15/1/23	800	700	850	Oil	—	No
16/1/23	600	480	600	Air	—	No
19/1/23	600	500	850	Oil	—	No
22/1/23	600	480	600	Air	—	No
23/1/23	600	500	850	Oil	—	No
23/1/23	600	500	600	Air	None	No

TABLE XI.

Record of Experimental Heat Treatment of Samples Nos. 8 and 9.

	°C.	°C.	°C.			
15/12/22	20	20	850	Oil	None	No
16/12/22	10	20	600	Air	—	No
18/12/22	600	480	855	Oil	None	No
19/12/22	20	20	600	Air	—	No
20/12/22	600	490	850	Oil	Yes	No
21/12/22	600	490	600	Air	Yes	No
3/1/23	600	500	850	Oil	Yes	No
5/1/23	600	480	600	Air	Yes	No
8/1/23	600	490	850	Oil	Yes	No
9/1/23	600	490	600	Air	Yes	No
10/1/23	600	490	850	Oil	Yes	No
11/1/23	600	490	600	Air	—	No
12/1/23	600	480	850	Oil	Yes	No
16/1/23	600	480	600	Air	Yes	No
20/1/23	600	490	850	Oil	—	No
22/1/23	600	480	600	Air	Yes	No

MANAGEMENT AND WORKING OF WATER-TUBE BOILERS IN MERCHANT SHIPS.

The Yarrow Boiler—The Babcock & Wilcox—The Belleville—The Niclaussé.

(*Concluded from page* 113.)

Before describing in detail the examination, &c., of the chief types of water-tube boilers, the small defects and repairs which may be dealt with by the engine-room staff, without outside assistance will be mentioned. The great point in running a number of boilers is to organise the boiler-room staff so that the small defects are taken in hand at once, also the hands must be instructed in their particular part of the work, especially those who are quite unskilled. It must be borne in mind that for economical working the casings and casing doors must be efficient, and as air-tight as possible, and the jointing material of all such doors must always be kept in good repair. Doors must be refaced and straightened as required, flame baffles and ash pans repaired, tubes straightened, and brickwork attended to immediately a defect occurs. In oil fire-boilers, it is usual, when a ship is undergoing a thorough refit, to lift part of the floor bricks in order to examine the framing and plating under. We will now consider particular types of water-tube boilers, but it must be borne in mind that the references to these types are more or less to the whole of the types in general use, and should therefore be considered as a whole.

The Yarrow Boiler.

This boiler is now the standard type for British war vessels, and is very often fitted in the mercantile marine. It has many excellent qualities, but the greatest from a marine engineer's point of view is its exceedingly simple construction. The boiler is made with medium and small-sized tubes, and lends itself particularly to oil burning. The earlier types were fitted with "D" lower water drums, but owing to wrapper plate trouble, *i.e.*, trouble with the longitudinal seams at the junction of the tube plates to lower drums, round lower drums were introduced. These round drums, although more expensive to manufacture and machine, appear to have solved the wrapper plate question. The Yarrow patent superheater has been fitted in many recent types. This superheater consists of a series of bent tubes expanded into headers, and is fitted on one side of the boiler only, where the width of the generating tube nest is reduced. To control the temperature of the steam an air-cooled damper is fitted in the funnel on the same side as the superheating element, the uptake being arranged for this purpose. The gases thus pass freely through the generating nest on the side opposite the superheater. The superheat gain should be about 100° Fahr. with steam at 260 lb. gauge. The automatic feed regulator may be either internal or external. The internal regulator consists of a balanced valve which controls the feed supply, and is actuated by a float, the float being pivoted at one end of a lever, a balance weight being fitted at the other end. Bearings suspend the lever, and a short arm on the lever is joined by a rod to a double-beat valve fitted horizontally. A cradle carries the lever bearings, the cradle being suspended in bearings fixed to the shell of steam drum. Attached to an arm on the cradle is a rod passing through a gland, and worked from outside the boiler, by means of which the position of the axis of the float lever can be altered, and the valve made to close earlier or later for the same position of the float. The box of the double-beat valve is jointed to the outside of the steam drum, the spindle passing through the shell. In the external type a float actuates a small valve which controls the movement of the feed check valves on the water drums of the boiler, thus regulating feed supply. In working this type of steam generator the fire rows of tubes must be constantly gauged for distortion, and a record made. Sudden distortion may be due to—

(*a*) Internal deposits, or grease in the feed water.
(*b*) The boiler being not free to expand and contract under the varying conditions.

Fire row tubes which fail should be plugged with the proper stoppers provided. On removal of a certain defective tube it will be necessary to cut out other tubes to obtain access to that particular tube, and the side should be selected which involves cutting out the minimum number of tubes. It is very necessary to keep the lower part of the steam drums free from dirt, ashes, &c. The removal and replacement of the tubes is a simple matter, and tubes so replaced should be noted in the boiler record.

After fires are down, the tubes must be thoroughly cleaned externally, and great care must be taken to ensure that there is no accumulation of soot or ashes on the water-pocket tube plates. Care must be taken to keep all casing joints quite tight, leakage of air causing rapid destruction, and no ashes or dirt must be allowed to accumulate in the casings. The holding-down bolts of the boiler must be so adjusted that the free movement of the feet is not restricted, and the bearings should be smooth and free.

Babcock & Wilcox Boiler.

The automatic feed regulator of this generator is very simple and efficient. It consists of a revolving piston valve actuated by a float. The water level may be adjusted by moving a piston valve lengthwise by an adjusting screw, an index being attached to the screw itself. Owing to the ports being cut diagonally in both piston valve and liner, the position of cut-off is altered by this operation. The free movements of the float can be ascertained by means of a handle attached to the regulator. As the regulator is solely intended to level the feed water when the boiler is steaming, it is necessary, when the boilers are shut off, to close the feed supply by using the hand-check valve. The handhole and manhole doors are jointed with wire-woven asbestos rings, and these rings must be black-leaded on one side only, *i.e.*, next to the header. It is essential that the surfaces of the headers are thoroughly cleaned before the handhole doors are put on. As regards the casing doors, special care must be taken that these doors are a good fit in their framing. All clips and catches must be efficient, and care must be taken that the asbestos packing between the headers is in good condition, in order to prevent cold air entering the boiler and causing a loss of draught with consequent drop in efficiency. In order to prevent the rapid burning away of casings and doors, it is necessary to keep the protective bricks in an efficient condition. In this type of generator corrosion is liable to occur in the upper bent rows of return tubes. This is due to the air pockets which form in the bends. After shutting off the boilers, pump up to a full glass, and when the steam pressure has fallen to 10 lb. per sq. in., open the air cock and pump the boiler right full from the feed tanks. On cleaning boilers, the return tubes need very special attention. A good method is to use a "sausage" made of canvas and stuffed. This is coated with blacklead and made a tight fit in the tubes. Insert from steam drum and pull to and fro by means of lines, until the interior surfaces of the tube are

[illegible] [illegible] [illegible] [

THERMAL AND MECHANICAL EFFICIENCIES IN STEAM AND MOTOR SHIPS I.

Comparative Heat Balances of Steam and Oil Installations—The Relative Net Efficiencies of Propulsion—The Impossibility of Steam Plants equalling the Efficiencies of Internal-Combustion Plants.

The competition between the oil engine and the steam engine is rapidly becoming extremely keen, and, in spite of a certain deep-seated prejudice in favour of the latter, due to long association and reliable service, the merit of the former as regards economy cannot be disputed, and whereas all the elements of time and further experience are on the side of the motor, in the case of the marine steam engine, improvements in efficiency are now only attainable with some difficulty. The interesting information which the North-East Coast Institution has obtained from the statements made in response to its questionnaire inviting comparative data from different builders of marine oil engines is capable of being expanded with great advantage, and an extension of the scope of the inquiry to cover the present-day economy of steam machinery would have added enormously to what is already a most useful review, albeit restricted to oil motors for one particular power only, and, unfortunately, not including by far the most widely-used engine of that type. Most interesting, perhaps, amongst the many figures brought forward were the anticipated consumptions, for these enable us to make certain deductions in regard to the relative efficiencies of the various types of engine. One of the most pungent criticisms of the motorship made in reference to the North-East Coast data had reference to the relative cost of an 8,000-ton 10-knot ship fitted with steam or internal-combustion machinery, and figures of £72,000 and £100,000 respectively were mentioned. Until further experience had been acquired with the latter, the shipowner, it was suggested, would require to allow 7½ per cent. for depreciation in the case of the motorship as against 5 per cent. for the steamship. The additional insurance of the extra £28,000 —40 per cent. more in first cost—was a further item, and additional upkeep had to be anticipated. These figures were suggested as being a formidable offset to the admittedly low consumption of the motorship.

The twin questions of how many heat units are required to drive a ship and where they go, cover very nearly every section of ship and engine design. To a very great extent, what they cost in bulk takes the matter of machinery out of the hands of the designer and leaves the choice of type of machinery to those who pay the fuel bills. It is interesting, therefore, at this juncture to compare the heat accounts of steamship and motorship with the object of determining the distribution of heat loss, of observing in what directions improvements are possible, and, perhaps, of ascertaining the range beyond which the superior utilisation of fuel heat must enforce the adoption of the internal-combustion motor in lieu of steam. At present the average best performances of steam machinery with air heaters, superheat and geared turbines show a consumption of nearly 1 lb. of oil per shaft horse-power per hour for all purposes. This implies a combined thermal efficiency of engine and boiler installation, with their essential auxiliaries, of only 14 per cent., and if the ratio of the effective power required to overcome the resistance of the hull to the shaft horse-power of the main engines be as much as 60 per cent., then the overall efficiency of heat utilisation is 8·5 per cent., a figure which is, if anything, well on the high side of usual steam practice. In a motorship of to-day the brake thermal efficiency of the engine is roughly 35 per cent., and if the hull and the propeller and shafting were the same in the two cases, this would give an efficiency of 21 per cent., or a fuel consumption of only 40·5 per cent. of that of the oil-burning steamship for the same speed and resistance. Compared with coal, the consumption of thermal units would be in this proportion too; the fuel *costs* of the two will depend on the calorific values of the fuels and the cost per ton—*i.e.*, in cost per B.Th.U. This comparison, of course, entirely ignores the other savings due to the use of oil, the gain in personnel, stowage, bunkering facilities, &c. We shall deal only with the dissipation of the fuel.

Comparative Heat Balances.

The thermal efficiency of propulsion is the eventual product of the thermal and mechanical efficiencies resulting from the numerous transformations of energy which the potential heat energy in the fuel undergoes between the furnace and the thrust block. The fact that in steamships this net value seldom rises above 7 or 8 per cent. must not be thought to mean that inefficiency is rampant throughout the system. Efficiency is a relative term, and a sequence of high efficiencies will end in a virtual zero if it be sufficiently prolonged. In what follows will be shown the altogether abnormal extent by which the thermal losses preponderate over the mechanical ones in the case of a steamship, and to what a relatively small extent are the mechanical losses responsible for the wide difference in fuel economy between steam and motorships. So much has been written of late years on the subject of propeller efficiency and the influence of position, &c., on performance, that the precise extent to which these factors influence the fuel bill has been very largely ignored of late, and, in point of fact, in the case of an efficient boiler and geared-turbine installation the sum of the mechanical losses due to the propeller, to hull efficiency and to transmission shafting, represents only about 5 to 6 per cent. of the total heat in the fuel used, whereas the sum of the thermal losses in the machinery may amount to from 83 to 88 per cent., or even more.

Considering first the case of the steamship, we will assume that the value of the heat energy in the fuel supplied represents 100. Propulsion is effected by means of a series of transformations of energy as follows :—

(1) The chemical reaction between the fuel in the furnace and the oxygen in the air supply, which results in the liberation of heat, may or may not be complete, and the furnace efficiency, albeit a constituent part of the total boiler efficiency, indicates the extent of the first source of loss of heat.

(2) The efficiency of the transmission of the heat in the product of combustion to the water in the boiler may be as high as 80 per cent. or more in special cases, but the quantity of heat which disappears immediately up the funnel and into the atmosphere is inevitably so considerable that a limit in total boiler efficiency of about 85 per cent. under the very best conditions of installation and operation may be accepted. In practice the average efficiency is very much below this, particularly when coal is used.

(3) In the piston engine it is the static expansion of the steam in the cylinder which causes the linear motion which is converted into rotary motion by means of connecting rod and crank. The losses involved in these processes are :—

(*a*) Condensation of steam on the relatively cool internal cylinder surfaces.
(*b*) Ineffective expansion in the clearance spaces.
(*c*) Injection of considerable percentage of heat to the condenser on release from low-pressure cylinder; and
(*d*) The mechanical friction of the engine.

[illegible] as these losses usually stand, it is when one comes to compare these sources of loss with what occurs in the internal-combustion engine cylinder that their magnitude is realised.

In the turbine engine there are two transformations of energy. In the fixed blades there is the transformation of the potential energy in the steam into kinetic energy, and in the moving blades kinetic energy is transformed into useful work. Neither is accomplished without material loss of heat, and the conversion of energy into useful work in the turbine is only accomplished at an efficiency which is low compared with that of the piston and cylinder. The superior economy of the turbine arises from the fact that it can utilise a much wider range of expansion at the exhaust end and a much higher initial temperature at the inlet end, and that, consequently, the available energy per lb. of steam is very much greater than it is in the case of the piston engine, which, for both thermal and mechanical reasons, cannot make use of the low vacua which are essential to enable the turbine to achieve its superiority. In the turbine, condensation and clearance losses are very small; on the other hand, leakage, exhaust velocity loss and steam friction in blade passages and on rotor are considerable, and the latter affect considerably the greater mechanical friction of metallic parts in contact which we find in the piston engine.

But the real trouble in both cases is the fact that as water (and subsequently steam) is utilised as the working fluid, and that as there must be the double change of state, first from water to steam and then from steam to water again prior to the future utilisation of the same fluid, an enormous loss of heat takes place in the exhaust and is carried away by the circulating water. So vast is this amount in percentage of the total heat used that it is actually from 40 to 50 per cent. greater than the total fuel heat supplied to the motorship to produce the same thrust. One source of heat loss alone, in fact, in the steamship represents more than 50 per cent. more than the total amount which the motorship requires. It is very doubtful if this fact is generally grasped. About 6 per cent. of the total heat in the fuel is used in overcoming the resistance of the form; no less than 60 per cent. at least is lost in the circulating water necessary to change the state of the working medium from steam to water, and hence to produce the vacuum. This loss is an inevitable one as long as steam is used, but its magnitude is often ignored. It is the real obstacle to obtaining very much better results than we get at present with steam. It is, as we shall see later, one of the reasons in the motorship why regeneration by means of the exhaust gases yields so little net result. Increased pressure and superheat will render the available heat drop greater, but to utilise it the size of the turbine must be increased as far as number of rows and blade velocity are concerned in the ratio of the square root of the heat drop. Moreover, increased pressure alone offers little or no advantage except for large powers or powers increased in more than the present ratio; it also means the abandonment of the cylindrical boiler.

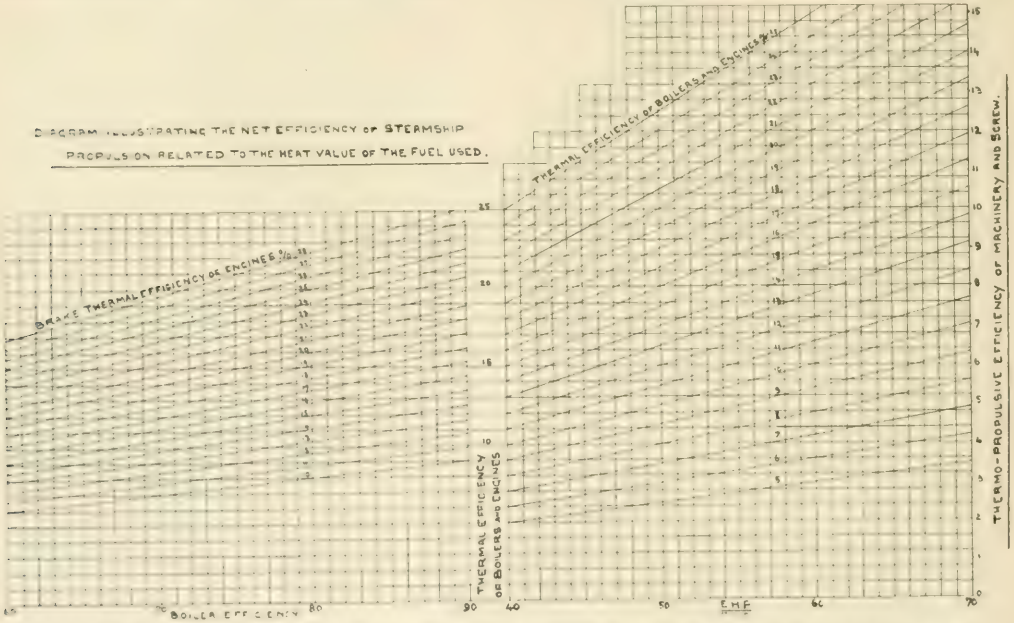


DIAGRAM ILLUSTRATING THE NET EFFICIENCY OF STEAMSHIP PROPULSION RELATED TO THE HEAT VALUE OF THE FUEL USED.
BRAKE THERMAL EFFICIENCY OF ENGINES
THERMAL EFFICIENCY OF BOILERS AND ENGINES
THERMO-PROPULSIVE EFFICIENCY OF MACHINERY AND SCREW.
THERMAL EFFICIENCY OF BOILERS AND ENGINES
BOILER EFFICIENCY
E.H.P.

Fig. 1.—Diagram connecting Engine and Boiler Thermal Efficiencies, showing Net Utilisation of Heat in Fuel.

The thermal efficiency of the boiler is the product of the furnace and the transmission efficiencies, and is—

$$\frac{\text{Heat produced in furnace per lb. of dry fuel}}{\text{Calorific value per lb. of dry fuel}} \times \frac{\text{Heat transmitted to steam per lb. of dry fuel}}{\text{Heat produced in furnace per lb. of dry fuel}} = \frac{\text{Heat transmitted to steam per lb. of dry fuel.}}{\text{Calorific value per lb. of dry fuel.}}$$

The thermal efficiency of the engine (or turbine) is the ratio of the heat equivalent of the horse-power developed (it may be indicated horse-power and indicated efficiency, or brake horse-power and brake efficiency) to the total heat per lb. of steam supplied, multiplied by the lbs. of steam used per horse-power developed.

The thermal efficiency ratio of the engine is the ratio of the heat equivalent of the horse-power developed to the available heat drop per lb. of steam, multiplied by the lbs. of steam used per horse-power developed.

The former measures the efficiency from the point of actual consumption, and the latter is the measure of the excellence attained within the limits of the cycle adopted, and one can either obtain the same ratio with an efficient machine working on a restricted cycle or an inefficient machine working on a equivalent of some 12 to 14 per cent. is left, and hence the loss of total heat in this region, due to a propulsive coefficient, is only some 6 to 7 per cent., although the horse-power loss is actually 50 per cent. The tables are self-explanatory, but in lieu of commencing with heat energy in fuel = 100, we may

E.H.P.	Propulsive Coefficient*		B.Th.U. supplied to shaft.	Thermal efficiency of boiler and engines combined.			Thermal efficiency of oil engine.	
				8·4	12·3	16·8	32	28
1	$\frac{1}{\cdot 58 \cdot \cdot 95 \cdot \cdot 80} = \frac{1}{0 \cdot 44}$	=	2·275	27·1	18·5	13·55	7·12	6·0
				↙	↙		↙	
1	$\frac{1}{\cdot 63} \cdot 1 \cdot 00 \cdot \frac{1}{\cdot 87} = \frac{1}{0 \cdot 548}$	=	1·825	21·7	14·84	10·86	5·7	4·8
				↙	↙		↙	
1	$\frac{1}{\cdot 68} \cdot \frac{1}{1 \cdot 05} \cdot \cdot 94 = 0 \cdot 672$	=	1·49	17·75	12·1	8·88	4·66	3·92

Total B.Th.U. required in fuel.

* From Table 1.

Tabular Representation of Variation of Heat Units required for Propulsion.

more extended cycle. This is what occurs in piston engine or turbine to a very large extent.

In the case of a steamship, which, assuming that we measure our power developed immediately abaft the thrust block and consider the thermal efficiency of engine and boiler combined, including the loss in piping between the two, this efficiency is merely—

$$\frac{\text{Heat equivalent of s.h.p. developed.}}{\text{Calorific value of fuel as fired} \times \text{lbs. of fuel per s.h.p. per hour.}}$$

Now, in a steamship this fraction may range from 8·5 to 17 per cent. as an extreme range. In a motorship it may run from 32 to 38 per cent., and we see at once the enormous loss in the thermal section compared with the mechanical section. This is very clearly indicated in Tables I and II.

In Table I, taking average efficiencies in the left-hand column and extremes in the two right-hand columns, it is seen that by the time the heat energy is transmitted into shaft horse-power somewhere between 83 and 92 per cent. of it has vanished, mainly out of the funnel and condenser outlet. In an average installation some 12 to 14 per cent. is supplied to the shaft in the form of power.

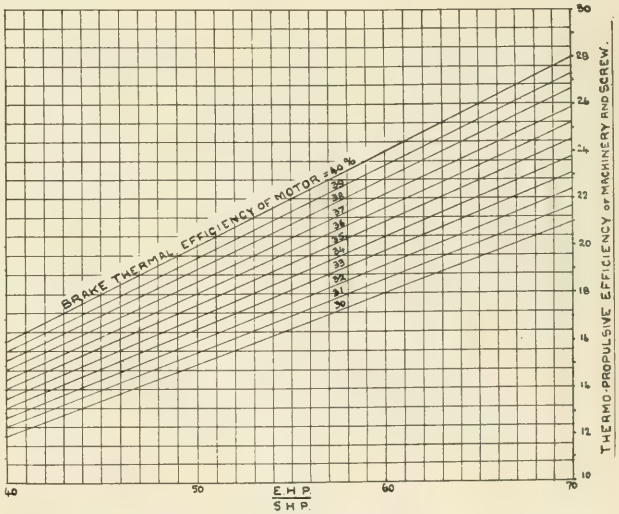


Fig. 2.—Oil Engine Brake Thermal Efficiency as in Fig. 1.

Mechanical Losses.

The mechanical losses between the torque in the shaft and the pressure on the thrust block in a given vessel are only a very small fraction of the total. The importance of attention to propeller design and position lies in the fact that it may enormously increase the total quantity of heat required to produce a given result. The three subdivisions of mechanical losses are: (*a*) the shafting, (*b*) the propeller position, and (*c*) the propeller itself. Their combined influence is usually considered as the propulsive coefficient or ratio of $\frac{\text{E.H.P.}}{\text{S.H.P.}}$ This value usually runs between 0·50 and 0·60, and it is very largely the absolute figure thus quoted that has blinded so many to the really small loss that occurs in the propeller region when once the screw is chosen. The much-maligned propeller, in point of fact, wastes horse-power and not heat. By the time the heat is converted into horse-power only the also proceed to consider the problem *ab initio*—that is, we will take the E.H.P. as unity in the average and extreme cases, and, using the thermal and mechanical efficiencies given in Table I, see how the total quantity of heat units required varies under the three conditions. The greater influence of thermal than of propulsive efficiency is very noticeable.

There are a number of lessons to be learnt from this distribution of efficiencies and losses as shown in Tables I and II. They provide material for deep reflection. In Fig. 1 is shown a diagram connecting engine and boiler thermal efficiencies with propulsive efficiencies in order to show the net utilisation of the heat in the fuel. Fig. 2 shows oil engine brake thermal efficiencies in the same manner. If it were possible to run a ship with the same brake thermal efficiency of turbine as that developed at Carville by the Parsons two-cylinder 10,000-kilowatt units, which was 25 per cent., and with the high boiler efficiency of 82·5 per cent. stated to have been exhibited by the R.M.S. *Saxonia* during the Boiler Committee trials,

TABLE I.—Distribution of Efficiencies and Losses for Steamship Installation.

Heat Units in Fuel Supplied = 100

Stage	Average efficiency.	Loss in Stage	Analysis of loss		Approximate range of efficiency	Corresponding range of loss in stage.
Boiler	77	23	Furnace	1·5	70—84	30—16
			Radiation	3·5		
			Funnel	18·0		
Engine	16	64·7	Radiation	0·7	12—20	61·6—67·2
			Internal friction	4·0		
			Rejected to condenser†	60·0		
Transmission, shaft bearings, etc.	12·3	87·7	Of which { Condenser .... 68·5 p.c. Funnel .... 20·5 p.c. Balance 11·0 p.c.		8·4—16·8	91·6—83·2
Transmission, shaft friction, thrust block, to hull	87	1·6	Friction of Bearings	0·6	80—94	1·6—4·0
			Steam tubes	1·0		
Propeller position, wake gain, thrust deduction	100	Nil	Wake gain ± Thrust deduction		95—105	—0·4—+0·8
Propeller	63	3·95	Real slip	2·0	58—68	2·7—5·3
			Friction and resistance of blades	1·9		
			Cavitation	0·05		
Mechanical section	54·8	5·55			44—67·2	4·7—5·5
Thermal η × mechanical η = thermo-propulsive efficiency	6·75	93·25			3·7—11·3	96·3—88·7

TABLE II. The corresponding figures for a motorship requiring the same quantity of net work, and using the same propellers, propeller position, shafting and form, result in the following heat balances :—

Heat units equal to net efficiency of steamship	6·75			3·7			11·3		
Transmission, hull and propeller efficiency as in steamship	54·8			44·0			67·2		
B.Th.U. supplied to shaft	12·3			8·4			16·8		
Brake thermal efficiency of engine	32	34	36	32	34	36	32	34	36
Heat units required in fuel	38·5	36·2	34·2	26·25	24·7	23·3	52·5	49·5	46·7
Actual loss in thermal section	26·2	23·9	21·9	17·85	16·8	14·9	35·7	32·7	29·9
Per cent. of loss of total heat in engine	68	66	64	68	66	64	68	66	64
Transmission	4·2	4·5	4·8	6·1	6·5	6·9	1·9	2·0	2·1
Hull efficiency	— 14·4 —	15·4 —	16·3	1·5 17·9	1·6 19·0	1·7 20·1	+ 1·5 10·5	1·6 11·2	+ 1·7 11·8
Propeller	10·2	10·9	11·5	10·3	10·9	11·6	10·1	10·8	11·4
Thermo-propulsive efficiency of engine and total heat supplied	17·6	18·6	19·7	14·1	15·0	15·9	21·5	22·8	24·2

the overall efficiency would be 20·625, and, given the high propulsive coefficient of 0·60, the thermo-propulsive efficiency of the system would be 12·4 per cent. Such a result (see Fig. 2) is obtainable with 31 per cent. brake thermal efficiency of oil engines in association with a propulsive efficiency of only 40 per cent., but as values of 38 × 60 respectively (= 22·4 per cent.) appear realisable, the hopelessness of attempting to perform with steam what can be done with the oil engine immediately becomes apparent.* The average motorship propulsive efficiencies are probably lower than those usually obtained in steamship practice on account of the lower pitch ratios, but, even so, inspection of these diagrams indicates that, even with considerable improvements over the best existing land practice in the way of boiler and engine efficiency, the steamship cannot begin to enter the range of motorship efficiency from the thermo-propulsive point of view.

(To be continued.)

* This statement, of course, refers to a special case, and then not to all branches of it. Where the scope of the two engines does clash, the motor is the more efficient. It cannot, of course, pretend to compete with steam in many applications, and in these the low efficiency is accepted as inevitable. Thus, in large central supply stations we find no oil engines, but for every ton of coal burnt the equivalent of some 12 cwt. disappears in the condenser cooling water.

ELECTRIC TRANSMISSION OF POWER FOR PROPELLING MACHINERY.

Mr. W. J. Belsey's Description of a Recent Application of the Diesel-Electric Drive.

The paper, read by Mr. W. J. Belsey before the Institution of Engineers and Shipbuilders in Scotland on April 10, dealt with the application of electricity as a means of transmitting power from prime mover to propeller. The author

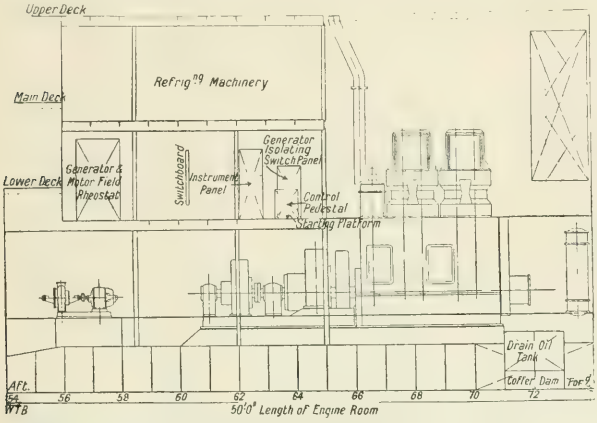


Fig. 1.—Machinery Arrangement, Diesel-Electric Drive, Merchant Ships, 3,000 h.p.

at the outset indicated that he would deal mainly with the turbo-electric and Diesel-electric systems in their application to merchant ships. He then proceeded to discuss the turbo-electric drive. Particular interest centres around his remarks on the Diesel-electric drive, especially as the author refers specifically to the three new fruit boats building by Cammell, Laird & Co., the first of which was launched last month.

The Diesel-Electric Drive.

Mr. Belsey pointed out that this method of propulsion has been comparatively little used up to the present, and has been confined to comparatively small ships of from 400 to 800 s.h.p. There are, however, he stated, three merchant ships of 3,000 h.p. building at the present moment in the country which are being fitted with this type of propelling equipment. Obviously, alternating current is the most suitable for turbo-electric drive, but for a Diesel-electric drive for smaller powers direct-current is attractive, principally on account of greater flexibility allowing constant speed of the engine irrespective of the speed of the propeller. With alternating current any speed variation in the speed of the propeller would have to be made by varying the speed of the engines, unless pole-changing on the motors was resorted to. Even then only two definite speeds could be obtained economically, and any intermediate speed would have to be obtained by varying the speed of the engine, and this variation in the speed of the main engine would obviously not be satisfac tory for driving the auxiliary generators. Alternating-current equipments have been designed for ships up to the highest powers, but as yet none of these designs has reached the stage of manufacture.

Following is a description of the equipment now being made. The machinery arrangement is shown in Figs. 1 and 2. Each equipment consists of four 750 b.h.p. Fullagar-Diesel engines running at 250 r.p.m., each direct-coupled to a 500-kw. 220-volt main generator and a 200-kw. 220-volt auxiliary generator. The main generators are connected in series, and drive a double-unit 2,500 h.p. motor, running at 90 r.p.m., direct-coupled to the propeller shaft. The auxiliary load, when all the refrigerating plant and the hold fans and auxiliaries are running, amounts to over 500 h.p. The connections are so arranged that any of the main generators can be used for auxiliary load when in port, say, for operating the cargo winches, &c.

Generators and Motors.

The main propelling generators do not require any special description. They are of standard type, only special care has to be exercised in the insulation of the machines, and the creepage surfaces have to be very much greater than those usually employed on land. Fig. 3 shows a section of a suitable direct-current generator of 500-kw. capacity. Beyond 1,500 h.p. it is generally more convenient to split the main motor into two units and couple the same together back by back. This provides a large factor of safety, as one-half of the motor can be cut out and the ship may run on the other half if desired or if necessary.

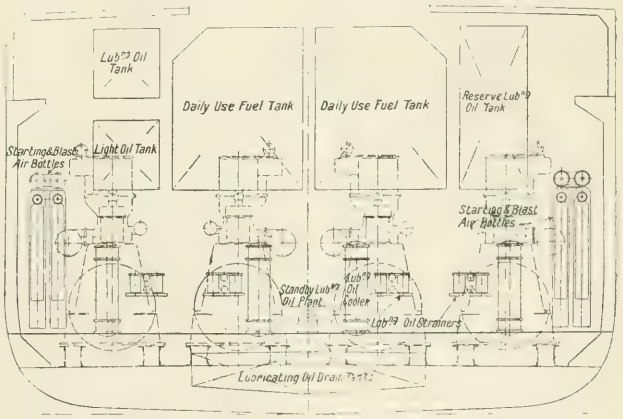


Fig. 2.—Machinery Arrangement, Diesel-Electric Drive, Merchant Ships, 3,000 h.p.

The motor would be designed with special reference to accessibility, so that all likely repairs can be carried out in the ship.

Diesel-Electric Control.

When dealing with direct-current propeller equipment of over 500 h.p. in capacity, a variable-voltage control appears

to give the best results, and this is employed in the case referred to. If, therefore, the field strength of the generators is reduced to zero the motor will stop, and if the generator field is built up in the opposite direction the motor will reverse in rotation. The main connections in this system are shown in diagrammatic form in Fig. 4, from which it will be seen that the four main propelling generators are connected in series, and each generator is provided with a potentiometer rheostat, the switch arms of which are all mechanically geared together and operated from the same way-shaft. It will be appreciated that, on an equipment of this size, the rheostats

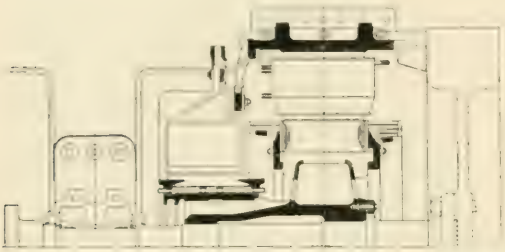


Fig. 3.—Section through D.C. Generator, 500 k.w. Capacity.

will be of considerable dimensions, and that to operate the switch arms quickly by hand would entail considerable physical effort. Therefore, the way-shaft is operated by a small motor, which is operated in turn by a master controller on the control platform, and the control gear is fitted with a follow-up device (similar to that used in steering gears), which cuts the motor out of circuit directly the rheostats are in the position indicated by the master-control lever. The master controller is of the reversing drum type, the drum being directly connected to the lever, but the body which holds the fingers is not held stationary; it is rotated by the follow-up device. By the side of the control lever are two pointers working in parallel slots, one being directly attached to the control-lever spindle and the other to the follow-up motion, so that, no matter how quickly the control lever is manipulated, the actual rate of acceleration or deceleration of the propeller motor is always dependent on the speed of the rheostat-operating motor. The quickest speed allowable can be found on trials, and, once set, the operator is not able to impose undue stresses or loads on the system.

In addition to the safeguards given to the system by the fixed speed of the rheostat motor, further protection is afforded by an overload relay and a speed-limiting switch on the engines. The overhead relay is fitted in the main power circuit, and whenever the current exceeds a predetermined value, either when accelerating or decelerating ahead or astern, it immediately interrupts the supply to the rheostat motor, and prevents the operator increasing the current beyond this point. It is so arranged, however, that if the relay interrupts the rheostat motor supply on an accelerating load, a movement of the control lever back towards the stop point automatically resets the relay and allows of a reduction in propeller speed being made. The overspeed switch is actuated by the engine governor, and is arranged to interrupt the power supply to the rheostat motor whenever the engine speed causes the fuel supply to be entirely cut off. If power is cut off from the propeller motor, it will continue to revolve in a forward direction due to the way on the ship. Therefore, if the motor fields are excited, a potential will be induced in the armature windings, and current will be delivered into the system.

Due to the fact that the power which can be dissipated in overcoming engine friction, and the auxiliary load is constant throughout the retarding period of the propeller, while the voltage of the electrical power returned by the propeller motor to the system is falling with the speed, it is obvious that when the propeller approaches zero speed the braking power current may require limiting. This is taken care of by the overload current relay, which functions equally well on an accelerating or decelerating load, and under any condition when the current circulating in the main power circuit exceeds a predetermined safe value the supply to the rheostat motor is cut off. Immediately the rheostat motor stops, all further increase in the braking load is stopped, and, due to the ship itself slowing up in the water, the braking load begins to fall

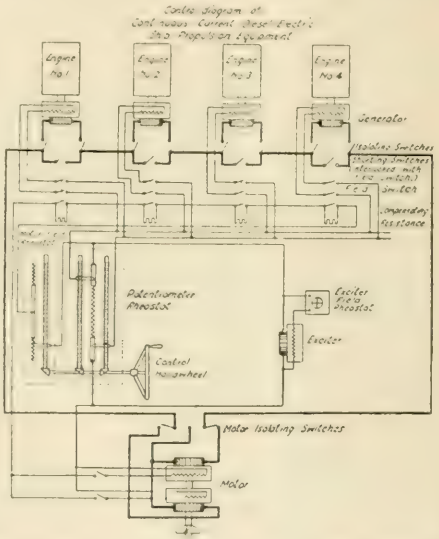


Fig. 4.—Main connections in D.C. Propeller Equipment of over 500 h.p. Capacity.

off. This allows the overload current relay to reset and complete the rheostat motor supply circuit. The result is that the system is simple to manipulate, and is equipped with all the safeguards necessary to prevent the operator abusing the equipment when manœuvring. Overload trips and similar electrical devices are not provided, it being considered undesirable to allow of the possibility of a sudden total interruption to the power supply such as might conceivably occur when manœuvring.

Bridge Control.

It should be borne in mind that marine conditions are very different from those prevailing on land, as in traction or even ordinary industrial work. At sea it is necessary to safeguard the vessel, whereas on land it generally suffices to allow the overload to bring about a complete shut-down of the electrical machinery. This becomes very evident whenever the question of bridge control is considered, a problem which naturally appeals to an electrical engineer as being fairly easily solved. The actual carrying out of a bridge control gear is admittedly not a difficult job, but for that matter the control of any machinery from a distance—remote control, as it is usually termed—is just as easy. Due to the peculiar conditions obtaining in an engine room at sea, up to the present, at any rate, it has not seemed a really desirable feature, and this is principally because of emergency conditions which make it necessary for the stressing of the machinery to be left entirely to the discretion of the engineer.

THE THAMES FERRIES "SQUIRES" AND "GORDON."

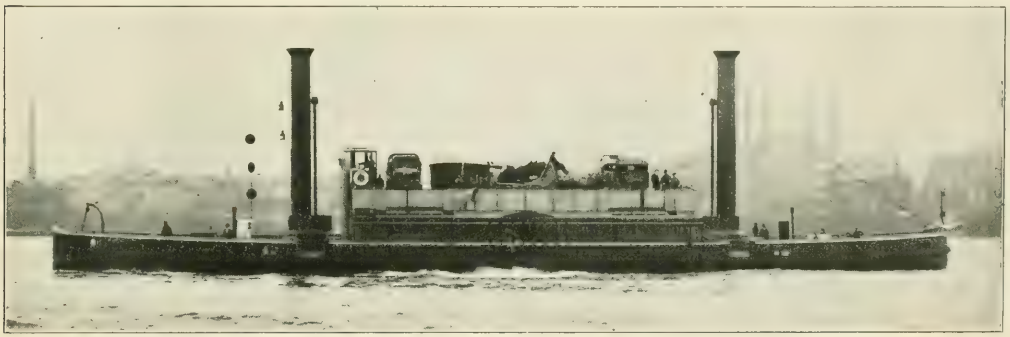
Some details of the Machinery of two New Woolwich Ferries built by J. Samuel White & Co. Ltd., Cowes.

Two ferries were constructed recently for the London County Council in order to maintain the free passenger and vehicular service between Woolwich and North Woolwich. These vessels are named respectively *Squires* and *Gordon* and they were built and engined by J. Samuel White & Co. Limited, East Cowes, Isle of Wight, in order to replace some older vessels of a similar type. The Woolwich ferries are interesting from a structural standpoint in that the vehicular deck is the top or superstructure deck, whilst passenger accommodation is provided on the main hull structure deck; this arrangement is necessary owing to the conditions of the river bank at either end of the crossing and the consequent construction of the landing stages. For this reason a rather peculiar arrangement of propelling machinery and boilers has been necessary, as will be seen from the description of the machinery. The principal characteristics of the two vessels are as follows:—

Length, b.p.	171 ft. 6 in.
Breadth, moulded	44 ft.
,, extreme	62 ft.
Depth, moulded	7 ft. 6 in.
Draught, loaded	4 ft. 6 in.
Speed (measured mile)	8·77 knots.
Number of passengers	1,000
Vehicular traffic	90 tons.

The main propelling machinery is situated amidships, and there are two boilers of the gunboat type arranged in separate boiler-rooms at each end of the machinery space. This arrangement is rendered necessary, as has been explained, by the special construction of the superstructure with the vehicle deck on the top. The auxiliaries, which are few and compact, are arranged in wing spaces on either side of the main machinery space as follows:—In the starboard wing space at the forward end is fitted a combined unit consisting of a centrifugal type circulating pump and a direct-acting beam-driven air pump. Both of these are driven by a common single-cylinder reciprocating engine of the Drysdale type. At the aft end of this space is situated a stand-by Weir feed pump; there are three feed pumps fitted in all, the other two being situated one in each stokehold. In the port wing space there is a duplication of arrangement with the exception that the reserve feed pump is replaced by a bilge or general service pump of Dawson & Downie make. On each side in the wing spaces is situated a feed-water tank, each tank containing its own filter.

The main engine-room contains the main engines and condensers only; these consist of two independent sets of simple condensing engines, each engine having two independent opposed and inclined cylinders, which have a diameter of 33 in., a stroke of 36 in., and a valve travel of 6 in., and which take steam from the boilers at 40 lb. per square inch. We illustrate on page 169 the aft cylinder of the port engine; the photograph shows also the reversing gear, which is of the direct-steam type. The engines are controlled, as is usual in such vessels, from a control platform on the upper deck, and this is illustrated also on page 169.



A Broadside View of the Woolwich Ferry "Gordon" in Midstream.

The boilers, as has been mentioned, are of the gunboat type, and they are arranged in two distinct boiler-rooms at each end of the machinery space. The boilers in the two vessels are approximately 9 ft. in diam. by 17 ft. long.

The working pressure of the boilers is 40 lb. per square inch, and the average coal consumption is about 8 tons of coal per working day of 20 hours. The coal bunkers, of which there are two, are arranged on alternate sides of the ship, and they carry each about 18 to 20 tons of coal, the total coal capacity being between 36 and 40 tons.

Speed trials were carried out on both vessels on the Stokes Bay measured mile, and a speed of about 8·77 knots was obtained, after which both vessels proceeded to the Thames under their own power.

The ships are lighted throughout by electric light, and the engine and dynamo for supplying this is fitted in the aft end of the control house; the deck machinery consists of two steam capstans, one fitted forward and one aft, for warping purposes, and there is a steam-steering gear, of direct-acting type, fitted in a compartment aft. We illustrate this on page 168. It will be observed that it is rod-controlled from the wheel on the bridge, and it appears to be a very quick-acting and efficient unit. A departure has been made in the *Gordon* and *Squires* from previous practice in the shape of the stern, which, whilst straight in profile, is paraboloidal in plan; the previous vessels were virtually double-ended.

AN INTERESTING DIESEL ENGINE DEVELOPMENT.

The Burmeister & Wain Double-Acting Four-Cycle Type for the New Swedish-American Liner.

An order was placed in March with Sir W. G. Armstrong, Whitworth & Co. Limited, to build a passenger liner for the Swedish American Line. Whilst the hull is to be constructed on the Tyne, the engines, which represent a very interesting departure and an important forward step in internal-combustion engine design, are to be built by Burmeister & Wain Limited; they will be constructed at Copenhagen, brought over to the Tyne for placing on board the new vessel. A twin-screw installation will be adopted.

The new machinery is an outcome of experimental work carried on for more than two years, whereby Burmeister & Wain have succeeded in carrying out the construction of an entirely new type of Diesel engine, namely, a double-acting four-cycle Diesel engine. The company have built an experimental engine of the double-acting type with a single cylinder, developing 1,000 b.h.p. This engine has been running periodically in their experimenting shop at Christianshaven for four months, and was in continuous operation for about one month, with very satisfactory results.

We illustrate this engine on page 170, and the photograph shows it on the test bed with the dynamometer attached. It is interesting to note that the largest marine Diesel engines of Burmeister & Wain type hitherto built have developed 300 b.h.p. per cylinder; the new engine develops 1,000 b.h.p. in the single cylinder. The 300-b.h.p.-per cylinder type has been used in combined passenger-cargo liner of between 13,000 and 14,000 tons d.w., having eight cylinder engines developing a collective brake-horse-power of 4,800.

The machinery for the new passenger liner for the Swedish American Line will consist of twin-screw engines, comprising two six-cylinder main Diesel engines of the new double-acting four-cycle type, developing a total of 13,500 b.h.p. during ordinary working at sea and giving the vessel a speed of 17 knots. There will be a complete set of auxiliary Diesel engines installed for operating the pumps, fans, &c., and for lighting purposes, consisting in all of six auxiliary Diesel-driven dynamos totalling 3,150 b.h.p. The new Burmeister & Wain double-acting marine Diesel engine is important in that it opens quite a new field in marine work for the use of the Diesel engine. The double-acting engine will make it possible to build Diesel engines developing the necessary horse-power for the propulsion of fast ocean-going liners at the high speed usually demanded, and this brings us within the range of classes of vessels from which the Diesel engine has been excluded hitherto because of its being unable to develop the required speed.

It should be noted also that Diesel motorships with the double-acting engines may be built as twin-screw ships in a similar manner to the steam reciprocating and turbine machinery plants now in existence; thus it will be possible to employ the same arrangement of the machinery without having to distribute the power over a larger number of shafts and propellers.

The new engine, besides being built for twin-screw installations, can also, in its long stroke form, be adapted to single-screw installations in cargo and passenger ships, which formerly have been provided with Burmeister & Wain's short-stroke Diesel engines; and it is considered that economy will thereby be effected. Diesel engines of the new type for single-screw ships can be built up to 5,000 and 6,000 b.h.p.

THE DANGER OF LEAKY FEED SYSTEMS.

One of the most important duties of the engineer or watch in a steamship is to see that a proper height of water is maintained in the boilers. When the feed pumps are working properly and the feed system is of adequate design, this is not a difficult matter, but if the pumps are not working satisfactorily or there are more or less severe leaks in the feed system, it may be a matter of great difficulty, if not indeed of impossibility, to maintain the correct height of water in the boiler, and immediate attention should therefore be paid to any such defects as soon as they develop. The penalty of not so doing is exemplified by the report of a boiler explosion which occurred on the coasting steamer *Kelburne*, which has been published recently by the Board of Trade. The boiler, which had been surveyed by Lloyd's, was in perfect condition, but the main feed check had cracked at the base of the flange where it was fixed to the boiler, and this crack had been repaired by means of a clip. It appears that the valve was leaking badly, necessitating an almost constant supply of extra feed, when one of the furnace crowns collapsed owing to overheating caused by the boiler being short of water. Happily no one was injured, but upon examination it was found that a new furnace crown was required, while the two top rows of tubes had to be renewed as they showed signs of being overheated. An examination of the feed check showed that the crack had extended about seven-eighths around the circumference, and it was leakage of the feed through this crack which undoubtedly led to shortage in the boiler. It is interesting to note that of the two combustion chambers, only one suffered damage.

ENGAGEMENTS FOR MAY.

Institute of Transport.—Fourth Annual Dinner of the Institute will take place at the Savoy Hotel, Strand, London, W.C., on Thursday, May 3. Ladies invited.

Institution of Civil Engineers.—Meeting at the Institution, Great George Street, Westminster, S.W., on Friday, May 4, at 6 p.m. "The Interdependence of Abstract Science and Engineering," by Sir Richard T. Glazebrook, K.C.B., F.R.S., Hon.M.Inst.C.E.

Institution of Mechanical Engineers.—Meeting at the Institution of Mechanical Engineers, Storey's Gate, Westminster, S.W., on Wednesday, May 2, at 8 p.m. Annual May lecture on "The Inner Structure of Alloys," by Dr. W. Rosenhain, F.R.S. Vice-President, of the National Physical Laboratory.

Iron and Steel Institute.—Annual meeting at Institution of Civil Engineers, Great George Street, Westminster, S.W., on Thursday and Friday, May 10 and 11, at 10 a.m. each day. Various papers to be read, followed by discussion. Annual Dinner in Grand Hall of the Connaught Rooms, Great Queen Street, London, W.C., on Thursday, May 10, at 7 for 7.30 p.m.

Royal Society of Arts.—Meeting at John Street, Adelphi, London, W.C.2, on Monday evenings, April 30, May 7 and 14, at 8 p.m. Three "Howard" Lectures on "The Development of the Steam Turbine," by Stanley S. Cook, B.A. (Cantab.), M.I.N.A., M.I.M. (Parsons Marine Steam Turbine Co. Ltd.).

Institution of Naval Architects. The summer meetings will be held in Holland from September 1 to 8 inclusive. The provisional programme has now been issued and particulars may be obtained from Mr. R. W. Dana, 5, Adelphi Terrace, London, W.C.2.

NOTES FROM THE FOREIGN PRESS.

Specially abstracted for "The Marine Engineer and Naval Architect."

Batch Drilling of Ship's Plates.

Experience with multiple ship production has tended to direct attention to the wider development of standard and multiple methods of ship construction. Although standard ships are now no longer built, there is no doubt that shipbuilders are definitely endeavouring to control the structural design of their vessels, such as by increasing and standardising the parallel middle body and by standardising rivet spacing. To best utilise these possibilities, either multiple punching machines or drilling machines are adopted. The latter have the advantage that batches of similar plates can be clamped together and thus much duplicate, laying-off and transport work is immediately eliminated. Schiess, of Dusseldorf, have given considerable attention to the development of drilling machines for shipbuilding purposes, and in this article their latest progress is outlined. Radial drilling machines are first described, the usual type consisting of two drills between two drilling tables. Batches of plates are placed on alternate tables, both drills concentrating on one batch whilst the batch on the other table is being made up. The drilling tables are fixed, and the spacing of the radial drill columns so arranged as to admit of the greatest length of plate likely to be met with being satisfactorily drilled. An improvement of the radial type of single drill is the portal multiple drill. This is best explained as being the conversion of the ordinary multiple punch. So far 32 spindles are fitted, and all can be controlled from the point of view of spacing and throwing out of gear. The batch of plates is fixed to a movable bedplate whose motion can be controlled on the self-acting principle. The machine is, of course, electrically driven and eliminates a considerable amount of labour. The smallest drill spacing is 60 mm. and the maximum drilling depth 120 mm. The machine can take a batch of six plates 10 m. by 2 m. by 25 mm. thick. Three different feeds are provided for the drills, 0·08, 0·12 and 0·18 mm. per revolution, the time taken to drill through 120 mm. varying from $1\frac{3}{4}$ to $2\frac{3}{4}$ minutes. Several of the machines have already been adopted in various German and other shipyards and, without exception, have worked well in practice.—*H. Becker, Schiffbau*, March 7, 14, 1923; p. 369; 6 pp.; 9 ff.

Practical Diagrams for Propeller Design.

In this article the author sets himself the task of presenting model propeller experiment data in the most convenient form for the practitioner. The outlooks of the experimenter and the practitioner are clearly distinguished. The various systems at present in use in Germany are first of all examined from the point of view of their physical dimensions. Having decided that any system should be composed of consistent metric units, the author directs attention to the most convenient rotational speed unit. He examines the suitability of pitch speed, *i.e.*, face-pitch times revolutions, tip speed, and finally a modification of tip speed, which equals diameter times revolutions, for the experimenter as well as the practitioner. He shows that the last expression is the most convenient, and this he substantiates by illustrating diagrams of thrust and torque constants and efficiency for each basis. The author next considers the various aspects of propeller design, and shows that these consist of four cases, in all of which the wake speed is known, but in two of which the thrust is fixed, and either the revolutions or the diameter; and in the remaining two cases the shaft torque is fixed, and also either the revolutions or the diameter. The nature of the fixing conditions determines the type of independent variable which will be most convenient for the practitioner. Diagrams in terms of the various independent variables are given, and form a convenient transformation of Schaffran's series A. The question of influence of disc area ratio is treated at some length, and several interesting features of its effect are brought out. For instance, when the value of the thrust constant, *i.e.*, $\frac{S}{\rho F (D n)^2}$ (in which S is thrust in Kg., ρ the density, F the disc area in m^2, D the diameter in m, and n the revolutions per sec.) proposed by the author equals 0·27, the efficiency is independent of disc area ratio. Decrease in disc area ratio gives, however, higher efficiencies for thrust constants less than this value, and increase in disc area ratio gives increased efficiency for thrust constants greater than this value. Finally, the various expressions used by the author are related to the usual units employed in shipbuilding, *i.e.*, the knot, the ton and the horse-power, and the necessary conversion factors indicated. The article forms a very complete treatment of the subject.—*Dr. Carl von den Steinen, Werft, Reederei, Hafen*, February 7, 1923, p. 61; 12 pp.; 11 ff.

The Construction of Nomograms by means of Logarithmic Paper.

In this article the author developes a rather hybrid set of nomograms. First, the ordinary alignment chart theory is developed, and then a curious transformation of the ordinary cartesian presentation of a logarithmic diagram is affected. This transformation enables the usual family of lines to be eliminated from the logarithmic diagram, but at the same time retains the disadvantage of a rectangular interpolator in conjunction with a diagonal support line. The awkward basis thus associated with the author's proposals naturally leads to complicated nomograms. Various examples are given, and deal principally with the Middendorf formula for resistance, the admiralty coefficient and the critical speed of propeller shafts. The author refers in his opening remarks to the practical utility of nomograms in practice, and suggests that the use of logarithmic paper will eliminate much of the difficulty of their construction. This, however, is doubtful, as a single logarithmic scale has been found generally to be quite convenient enough. The methods proposed by the author are not simple either to construct charts, or to utilise them. It seems that the artifice of a double scale, *i.e.*, such as a draught-deadweight scale, as supplied to ship's captains, is being curiously avoided.—*W. Kretschmer, Werft, Reederei, Hafen*, January 22, 1923, p. 31; $4\frac{1}{2}$ pp.; 5 ff.

Influence of Waves on a Ship in Motion.

In this article the general properties and behaviour of trochoidal waves are reviewed. The origin of waves—deep sea waves, shallow water waves—are briefly referred to. The chief portion of the article deals with the relation between the length, height and speed of deep sea waves. As is well known, the phenomenon of a moving wave is not explained by the bodily forward motion of the water, but merely by the orbital motion of the water. This orbital motion, however, is equivalent to translational motion at the crest and trough of the wave. At the crest this motion is opposed to the motion of the wave, and at the trough it is in the same direction as the wave motion. The speed of the wave is between six and seven times the translational speed at the crest and trough.

In observing wave properties at sea, the speed and direction of the ship relative to the waves must, of course, be taken into account. The author shows how the necessary correction is affected. A knowledge of the dimensions of waves likely to be met with on different routes can be utilised to ensure the provision of requisite strength and stability in any given vessel. Due to the passage of a wave when the wave crest is amidships, the vessel is lifted bodily, and when the trough is amidships the vessel drops bodily. In one case these vertical accelerations were measured by Mottez, a French captain, and he found that the ratio of apparent weight reduction on the crest to the apparent increase in the trough was $\frac{2}{3}$, the wave height being 26 feet. From this information it is possible to estimate the length of the wave, and this worked out at 408 feet. By calculating the total weight diminution when the crest is amidships, a knowledge of the wave waterline tons per inch enables the diminution of draught to be obtained. This reduction of draught may have a considerable influence upon the initial stability of the ship, especially when the metacentric height in the still water condition is small. (*To be continued.*)—*M. Teerling, Het Schip*, April 6, 1923, p. 82; 3 pp.; 4 ff.

Leader Cables.

The value of leader cables for the safe navigation of difficult harbour entrances is being recognised increasingly by all maritime nations. Each nation claims to have invented the device, and in this article the author, naturally enough, illustrates the important developments made to the subject by German scientists. The principle of the leader cable, as is well known, consists of a cable placed along the estuary bed of sufficient length to have its sea end well out of the shallows. From the shore end of the cable an alternating current is sent along the cable, the usual frequency being 50 per sec. for long cables, up to 500 for short ones. The magnetic field set up round the cable enables a vessel fitted with instruments for measuring the strength of, or detecting, this magnetic field, to determine its locality with respect to the cable. Usually instruments are fitted on each side of the ship, and thus the relative locality of the cable can be obtained by noting the relative strengths indicated by each instrument. These instruments indicate, either by sound or by pointer, the latter, of course, being the more convenient. The instruments generally used are described in principle in this article, and the technical considerations governing appropriate current strength and frequency are dealt with. The chief difficulty associated with the use of the leader cable is that of getting on to the cable initially. Research is apparently being made at present in Germany to test the efficacy of having a sound signalling apparatus at the cable end. Appropriate receivers on board ought, therefore, to locate the cable end without much difficulty. The author enumerates the more obvious advantages of the leader cable, but, in addition, points out its military advantage of controlling traffic through a mine field.—*Dr. F. Seidel, Werft, Reederei, Hafen*, April 7, 1923, p. 161; 3 pp.; 9 ff.

Economic Sailing Ships.

Previous to the war, when but little difficulty was experienced in getting full cargoes even for merchant vessels of the largest size, the economic efficiency of the steamer was unquestionably greater than that of the sailing ship. This fact was evidenced by the decline in the number of sailing vessels under construction. At the present time the economic position of the freight market is very different. Only on rare occasions can a full cargo be secured by ordinary sized merchant vessels. As the size of the ship is definitely fixed economically by the cargo it is likely to secure, present conditions seem to be producing a reaction in favour of the small sailing ship. To tempt a company to engage in the carrying trade, the initial outlay involved in the purchase of its fleet must be such that an adequate return on the capital is clearly forthcoming. With the present high price of steel, the return to wood sailing vessels appears profitable. From the above generalisations the author shows that the present reaction in Germany towards the small wood sailing vessel has some economic justification. In particular he describes the ss. *Desag*, a small three-masted schooner of 230 tons deadweight, which has worked exceedingly well in the Baltic. The author touches upon the question of the auxiliary motor. He points out that for the size of vessel under consideration the fitting of an adequate motor would involve an increase in initial outlay of almost 33 per cent. For the coasting work for which the vessels are intended the motor is probably an unjustifiable expense, whose advantage does not outweigh the advantage of simply securing a tug, when necessary, for harbour work.—*Oberingenieur Steiner, Schiffbau*, March 28, p. 415; 3 pp.; 4 ff.

Systematic Research on Air Screws.

In our previous number we outlined Dr. Schaffran's research programme on air screws, as published by him in *Schiffbau*, of February 14, 1913. In the subsequent article now under review, Dr. Schaffran discusses the presentation of the research results. A typical diagram only is given, and refers to the two-bladed screw of blade width ratio 7·5 per cent., and of nominal pitch ratio 0·80. As a speed base the quantities $\frac{V}{nD}$ and $\frac{V}{\pi nD}$ are chosen, the former following Dr. Schaffran's usual practice for air screws, and the latter being the practice of the Göttingen Air Experiment Station. The thrust and torque constants chosen are $C_1 = \frac{S}{\rho n^2 D^4}$ (S = Thrust), and $C_2 = \frac{Q}{\rho n^2 D^5}$ (Q = torque). All expressions are in metre-second kilogram units. This method of plotting brings out clearly the relation of face to effective pitch. This ratio in the example given was 1·08. The general character of the thrust and torque constant curves are discussed. The ideal efficiency curve is plotted with the actual, and the conclusion drawn that no revolutionary improvement in propeller efficiency seems at present attainable. In the remainder of the article Dr. Schaffran outlines the remainder of the work undertaken. Full particulars are not given, but these are apparently to be published in a forthcoming work of the author's, to be entitled *Systematische Luftpropellerversuche* (II Teil).—*Dr. Schaffran, Schiffbau*, March 21, 1923, p. 399; 2½ pp.; 1ff.

The Safety of Transatlantic Passenger Vessels.

This article forms an interesting review of the subdivision question generally, and gives a clear exposition of the safety of the Italian liner *Giulio Cesare* in particular. In his introductory remarks the author shows that the importance of subdivision has been realised for centuries, and quotes Nicolo di Conti, who, having voyaged to India in 1424 to 1429, wrote of the value of bulkheads in his ship. The now classical marine disasters which resulted in the London Conference of 1914 are referred to, after which the risks of navigation, as compared with land transport, are reviewed. The author quotes various English authorities to show that marine casualties have decreased to such an extent in the last thirty years that now the ordinary yearly land transport casualties of the U.S.A. exceed the whole yearly marine casualties. The influence of excessive safety precautions upon the commercial success of the marine passenger trade is referred to, and the findings of the 1914 Conference and the modifications of the 1921 and 1922 conferences examined in this light. The views of the various countries as to what should stand for the most satisfactory criterion of service are outlined and considered, particularly in the light of suitability of application to existing Italian vessels. The subdivision regulations themselves are next discussed, and the factors of subdivision, as proposed by the 1914 and 1922 conferences, together with the ordinary requirements of the classification societies, are dealt with.

Finally, the *Giulio Cesare* is examined in detail. Flooding curves of the vessel are given. As constructed, she is much safer than the most stringent requirements of the 1914 proposals. Actually she has 16 bulkheads. Her length is 631 ft. and the factor of subdivision thus required would be 0·40. Actually the factor of subdivision is 0·25, which is less than the minimum proposed, which refers to a vessel 932 ft. in length and is 0·33. The 1914 proposals necessitate 11 bulkheads. The 1922 proposals necessitate only 9 bulkheads, with a corresponding factor of subdivision of 0·45. It is interesting to note that the classification societies' requirements also necessitate 9 bulkheads. The safety of the *Giulio Cesare*, from the point of view of the 1922 proposals is thus very evident. Diagrams showing the trim of the vessel with four consecutive compartments flooded progressively along the length are given, together with tables showing the corresponding floodable lengths, amount of water in compartments, freeboard, and metacentric height after flooding. This latter varies from 0·792 metre to 0·335. Watertight doors and provision against fire are also referred to.—*N. Albini, La Marina Italiana*, February 11, 1923, p. 35; 7 pp.; 5 ff.

The Tangential Speed Component in a Propeller Race.

For a number of years attempts have been made by various devices to increase propeller efficiency by abstracting from the propeller stream its otherwise wasted rotational energy. Principal amongst these devices is the contra propeller. The fact that increase of efficiency has been definitely obtained by fitting a contra propeller directs attention to the magnitude of the tangential component. A study of the vortex theory of propellers is useful in this connection. From observation it is known that definite vortices release themselves at the trailing edge of a propeller blade, but principally from the tip and the base. The same phenomenon is evident in aerofoils of finite aspect ratio. For purpose of simplicity these helical vortices can be imagined as formed of a straight vortex and a ring vortex. With this simplification, and by applying the Joukowski circulation theory, the component velocities in the propeller race may be obtained. An example is worked out for the case of the *Imperator's* propeller, and a mean tangential speed of 2·1 metres per sec. was obtained. The particulars of the *Imperator's* screw are as follows:—Ship speed 23·6 knots, s.h.p. 18,750, r.p.h. 172, real slip 35·6, diameter 5.08 m., pitch 4·42 to of blades 4. A further investigation by the simpler and more approximate turbine blade theory gave 3·4 m. per sec. as the tangential speed. The correct value of this component can only be determined by experiment. However, the theory can be improved in the light of further experience, and thus a satisfactory basis for the design of contra propellers laid down.—*Dr. G. Bauer, Werft, Reederei, Hafen*, March 22, 1923, p. 146; 2 pp.; 5 ff.

RECENT PATENTS RELATING TO MARINE ENGINEERING.

Compiled by E. P. Alexander & Sons, Chartered Patent Agents, 306, High Holborn, London, W.C.

No. 193,117. Steam Boilers.

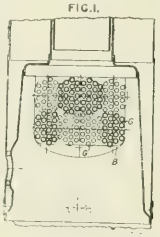
Taylor, E., Lark Hill, Bromley Cross, near Bolton. November 12, 1921, *No.* 30,202. [*Class* 123 (ii).]

Settings.—Relates to the settings for Lancashire, Cornish and like boilers comprising box-like metal casings *f* open at their front edges and built vertically into flue walls at the rear end *h* of the boiler, the boxes containing vertical plates *a*, the outer edges of which carry suitable packing *j* which bears against the end of the boiler, and consists in forming the plates *a* with inclined planes *c* at their rear edges *b* near the upper and lower ends thereof, the inclined planes being adapted to rest against horizontal rollers *d* carried by pins or studs *e* mounted in the casings. The plates *a* may be moved or withdrawn by hand by means of handles *k*.

No. 193,182. Steam Generators.

Kirke, P. St. G., Parliament Mansions, Victoria Street, Westminster, December 1, 1921, *No.* 32,279. [*Class* 123 (ii).]

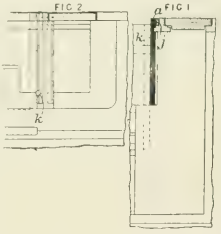
In a vertical boiler having parallel cross water-tubes G arranged within an extension of a cylindrical fire-box, the heating surface of the tubes is greater than 1·6 times the heating surface of the fire-box owing to a suitable arrangement of the manholes B giving access to the tubes. Two or more rows of manholes are provided opposite each end of the tubes. The major axes of the manholes of adjacent rows may be alternately vertical and horizontal, and the major axes of the manholes of one row at one end may be at right angles to the major axes of the manholes of the row at the other end.



No. 193,204. Steam Generators.

Hurd, M., 3, *Linden Avenue, Woodseats,* and *Milner Hurd's Patent Company Limited*, 4, *Paradise Square,* both in *Sheffield. December* 20, 1921, *No.* 34,274. [*Class* 123 (ii).]

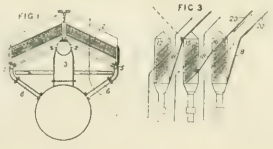
A gas-tight joint at the back of a Cornish or like boiler is formed by (1) steel or like transverse strips *a*, so supported as to rest at an angle of 45 deg. against the boiler end-plate at the top of the flues, and (2) asbestos or like ropes *k* suspended from the steel strips between the boiler end-plate and the bevelled ends of the side flue walls. Resilient insulating material *j* is placed over the steel strips.



No. 193,368. Surface Condensers.

Aktiebolaget Ljungströms Angturbin, 2, *Arsenalsgatan, Stockholm. May* 1, 1922, *No.* 12,216. *Convention date, February* 17. *Not yet accepted. Abridged as open to inspection under Section* 91 *of the Act.* [*Class* 32.]

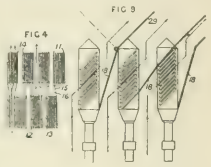
In an air-cooled condenser for locomotives, &c., the condensing elements consist of tubes, preferably flattened, connected in groups to headers themselves connected to a main collecting pipe. Baffles for directing the flow of air over the tubes are also provided. The elements 2 are arranged in an inclined position above a receiver 1 for steam and condensate, whence the steam passes to the elements 2 by pipes 3 and to which the condensate is returned by collecting pipes 5 and pipes 6. Fig. 3 shows three groups of flat, transversely inclined cooling elements connected to the headers 12, 15 and 16 which are connected to the collecting pipe 5. Baffles 18 are arranged as shown, and their upper parts 20 may be hinged so as to be moved, either by the driver or by wind pressure, into such positions that air is drawn through the cooler in the direction of the arrows, whether the locomotive is moving backwards or forwards.



No. 193,367. Air-Cooled Condensers.

Aktiebolaget Ljungströms Angturbin, 2, *Arsenalsgatan, Stockholm. May* 1, 1922, *No.* 12,213. *Convention date, February* 17. *Not yet accepted. Abridged as open to inspection under Section* 91 *of the Act.* [*Classes* 32 *and* 64 (iii).]

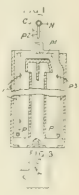
In an air-cooled condenser, especially for locomotives, in which the steam is condensed in a number of flattened tubes or like elements, the elements are arranged in groups, which are so arranged and to which the air is so directed by baffles, &c., that the cross-section of a group exposed to the air current is greater that the space between two groups through which the air-current enters or leaves the condenser, space being thus saved. For this purpose the tubes may be arranged in two rows as shown in Fig. 4, the air passing through spaces 12 between the groups 13 of the elements of the first row being guided by partitions 15, 16 among the elements of the second row, and the air passing through the groups 13 of the first row being guided by the same partitions into spaces 14 between the groups 11 of the second. Other arrangements, including one with three rows, are described. Fig. 9 shows an arrangement in which the air passes obliquely to the condenser, the elements being placed obliquely to correspond and the air being directed by baffles 18 as shown. The continuation 29 of the baffles 18 may be hinged and directed to suck air through the condenser by the motion of the locomotive whether it is moving forwards or backwards.



No. 193,315. Furnaces.

Parkinson, J., 1, *School Green, Thornton, Yorkshire. April* 11, 1922. *No.* 10,269. [*Class* 51 (i).]

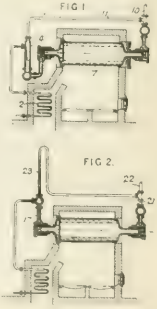
Steam is supplied to pipes P having nozzles P^3 in the side-flues of a steam generator or like furnace to consume smoke. In the rear flue the pipes P are connected by a pipe P^1 which has a branch P^2 leading to a rose N in the chimney C. The pipes P may be omitted when the flues are small and the steam is supplied direct to the pipe P^2, which has jets discharging into the chimney flue. When there are two boilers side by side, three pipes P are used.



No. 193,417. Steam Boilers.

Aktiebolaget Atmos, Lastmakaregatan, Stockholm.—(*Assignees of Blomquist, J. V. and Wessblad, K. F.*), *February* 16, 1923, *No.* 4,623. *Convention date, February* 20, 1922. *Not yet accepted. Abridged as open to inspection under Section* 91 *of the Act.* [*Class* 123 (ii).]

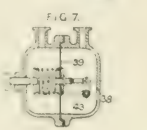
Steam generated in the feed-water heater of a rotary boiler is separated and led directly to the steam supply pipe, thus ensuring an undisturbed supply of water to the boiler. Hot water from a feed heater 2, Fig. 1, is supplied to the rotary boiler 7 through a vertical separating chamber 4 connected to the steam supply pipe 10 by a pipe 11. The feed pipe may open into a supply chamber 17, Fig. 2, connected to the steam supply pipe 22 by a pipe 23 having a U-shaped bend. A valve 21 maintains the pressure in the boiler above that in the steam supply pipe.



No. 193,418. Steam Boilers.

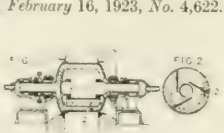
[illegible] *Ingeniörsaktiebolaget Atmos, Ingenjörsvägen, Stockholm.* Assignees of [illegible] and Wessblad, K. F., *February* 16, 1923, *No.* 4,624. *Convention date, February* 20, 1922. *Not yet accepted. Abridged as open to inspection under Section* 91 *of the Act.* [*Classes* 123 (i) and 123 (ii).]

The depth of the annular body of water in a rotary boiler 1, Fig. 1, is indicated by a gauge 4 connected to the water supply and steam outlet pipes. The gauge may consist of two vertical tubes 7, Fig. 2, dipping into a chamber 5 containing mercury, the top of the pipes being connected to the boiler by water filled pipes. In modifications the tubes are replaced by concentric chambers, and a float in one of the chambers carries a pointer visible through a gauge glass. In another modification, a chamber 9, Fig. 3, partly filled with water and in communication with the boiler through connections 10, 11, contains a float 12 having a stem formed with a valve 14, which closes the communication with the gauge glass 13 around the pointer when the glass breaks. The float may be connected to the pointer by a pivoted arm. The gauge chamber 38, Fig. 7, is connected to the boiler on opposite sides of a diaphragm 39 connected to a pointer 43.



No. 193,416. Steam Boilers.

Aktiebolaget Atmos, Ingenjörsvägen, Stockholm.—(*Assignees of Blomquist, J. V. and Wessblad, K. F.*). *February* 16, 1923, *No.* 4,622. *Convention date, February* 20, 1922. *Not yet accepted. Abridged as open to inspection under Section* 91 *of the Act.* [*Class* 123 (ii).]

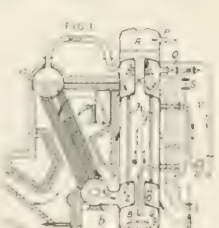


In a rotary boiler, one or more plates 2 secured to the boiler shell 1 are so arranged that during rotation of the boiler they distribute the water evenly over the inner surface of the shell. The shell may be fixed and the water rotated and distributed by rotating blades.

No. 193,426. Steam Boilers.

Winands, A., 35, *Wielandstrasse, Magdeburg, Germany. February* 19, 1923, *No.* 4,831. *Convention date, February* 20, 1922. *Not yet accepted. Abridged as open to inspection under Section* 91 *of the Act.* [*Class* 123 (ii).]

The upper and lower drums *a*, *b*, of a water-tube boiler are connected to two vertical drums *f* serving as feed-water heaters and as hot-water reservoirs from which low-pressure steam may be taken for heating purposes. Each vertical drum is divided by transverse partitions into five compartments A, X, *y*, Z, B. The compartments A, *y*, B are in communication with one another through tubes K, O. Feed-water is supplied to the compartment B and is heated by the furnace gasses and by the hot water in the boiler downcomer tubes *h* traversing the compartment *y* and connecting the compartments X, Z, which are in communication with the boiler drums. Boiler steam is taken off through a pipe Q, excess steam passing through a valve S into a pipe opening into the compartment B. Water is drawn from the compartment *y* and forced by a pump through a pipe *v* opening above trays fitted in the steam space of the compartment X. Steam is taken off from the hot feed-water through an outlet P.



No. 193,777. Steam Turbines.

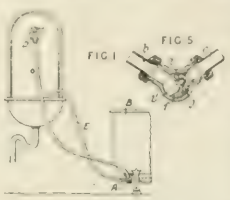
Aktiebolaget Ljungströmas Ångturbin, 2, Ansvarsgatan, Stockholm. July 10, 1922, *No.* 18,907. *Convention date, April* 18. [*Class* 110 (iii).]

A blade is formed with a packing edge projecting from the free end of the blade at the exhaust side and extending over a part only of the length and a part only of the width.

No. 193,716. Ejectors.

Delattre, J. H., 15, *Rue Hegesippe Moreau, Paris.* March 22, 1922, *No.* 8,335. [*Classes* 71 *and* 135.]

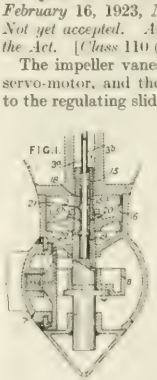
A device for filling and emptying receptacles such as baths is formed as a threeway rotary plug cock *f* operable by a handle A and connected to the water supply D, the discharge pipe *c* and the receptacle B. In the discharge position of the valve, as shown in Fig. 5, the pressure water supplied through the connection *b* flows through an ejector nozzle *j* and discharges the waste water communicating with the passage in front of the nozzle. The pipe *c* is in the form of a Venturi double cone at the part adjacent to the casing of the valve. The nozzle is formed of two narrowing cones i^2, *i* and a short cylindrical piece *h*. In the mid-position, the valve closes the discharge and supply pipes, while in the other extreme position water is permitted to flow freely into the bath, the discharge pipe *c* then being cut off. For the purpose of discharging water to a higher level, air may be admitted to the water in the discharge pipe E, to lessen the weight of the column of liquid being discharged, by forming this pipe at a suitable position with an adjustable air inlet.



No. 193,414. Turbines.

Englesson, J. E., 2, *Nya Kungsgatan, Kristinehamn, Sweden. February* 16, 1923, *No.* 4,607. *Convention date, February* 18, 1922. *Not yet accepted. Abridged as open to inspection under Section* 91 *of the Act.* [*Class* 110 (iii).]

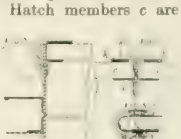
The impeller vanes 7 of a turbine are adjusted by a hydraulic servo-motor, and the pressure fluid for the servo-motor is supplied to the regulating slide 15 through a tubular coupling rod 3^a arranged in a tubular bore 3^b of the shaft. The servo-motor consists of a piston 16 on a tubular piston rod 3, the bore of which contains the cylindrical slide 15. The slide has two external annular grooves 15^a, 15^b which may be brought into communication with channels 20, 21, 22 in the piston rod 3. If the slide 15 is raised, the pressure fluid runs from the hollow rod 3^a through the channel 22 below the piston 16, thus raising the piston. The liquid above the piston runs through the channel 21 annular space 15^b and channels 20, 19 to the space 3^b. The piston 16 is connected by the piston rod 3 and cross head 8 to the vanes 7 so that the vanes are adjusted by the movement of the piston. When the slide 15 is moved downwards the pressure fluid passes through the channels 18, 21 to the space above the piston 16, thus depressing the piston to operate the vanes. The liquid beneath the piston escapes through the channels 22, 15^b, 20, 19.



No. 193,669. Ships' Hatches.

Le Brun, A. J., 37, *Ashdale Road, Mossley Hill, Liverpool. February* 18, 1922, *No.* 4,804. [*Class* 113 (i).]

Hatch members *c* are locked by plates *f* hinged to the hatch covering *a* and taking over the ends of the hatches, when in operative position, where they are secured by cleats *h* taking over hasps *j*. When the hatches are sectional and abut on athwart ship beam *e*, Fig. 7, the securing plates *k* are mounted on hinges *l* carried by the beam.



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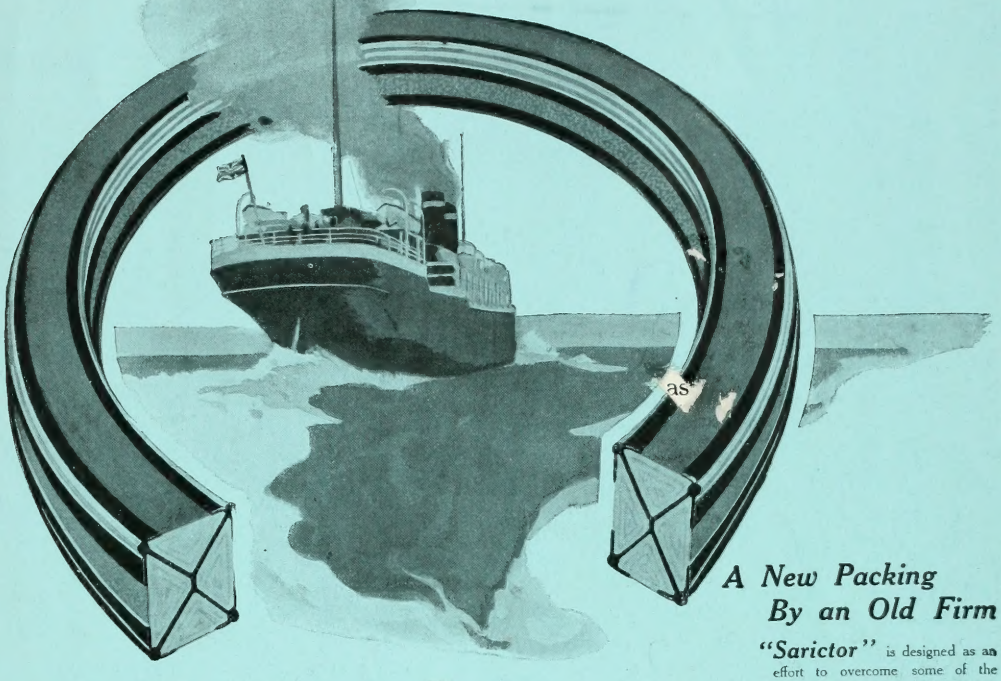

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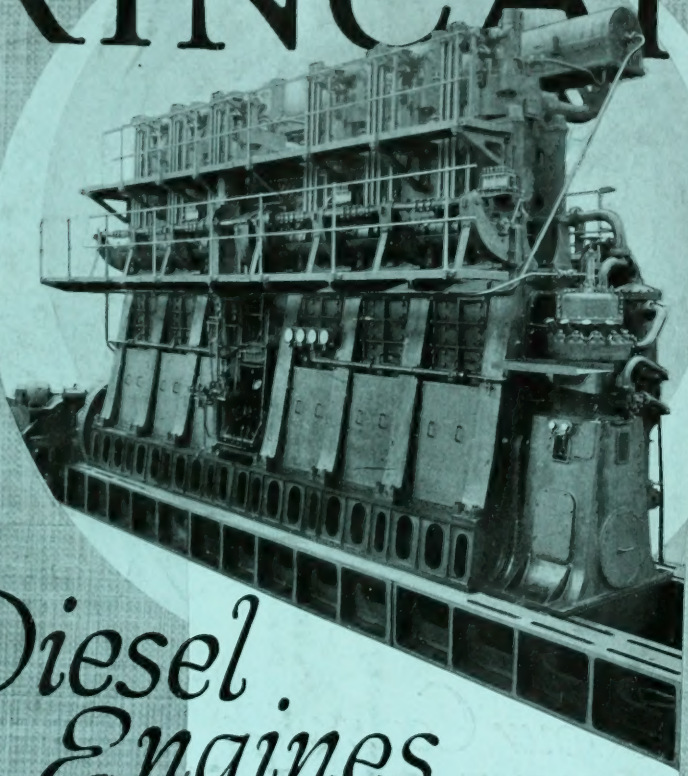

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